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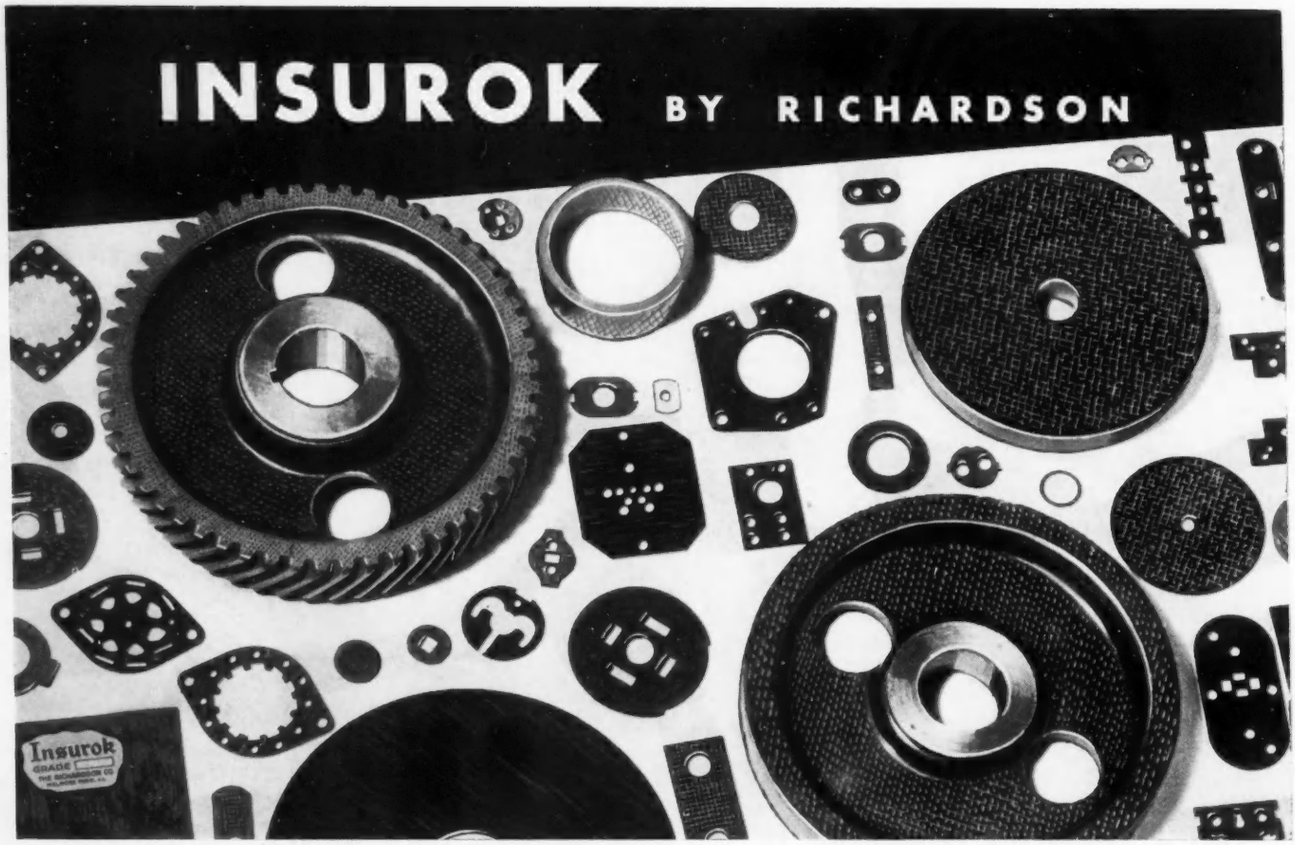

AUTOMOTIVE engineers for years have been looking for a new and better material for distributor heads, generator collars, timing gears, small gaskets, instrument panel cut-outs, horn button bases, stampings, washers of all kinds. In 1932 they made a discovery . . . INSUROK—by Richardson! A material that is stronger, tougher, yet more resilient. Shock-proof, strain-proof. Non-corrosive—will not soak up oil. Low in heat conductivity, high in electric insulation.

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A $700 Airplane?

What the Aeronautics Branch of the Department of Commerce is Doing to Foster It, with the Opinions of Some Designers on its Practicability

THOMAS R. MARSHALL, vice-president under Woodrow Wilson, is reported to have said that what this country really needs is a good five-cent cigar. Eugene L. Vidal, director of aeronautics in the present administration has said with thundering emphasis: what this country needs is a good $700 airplane. Judging by the results to date, Mr. Vidal's project has aroused the same flood of man-in-the-street response as did Mr. Marshall's aphorism.

It seems reasonable to assume, however, that the voice of the engineers who will be concerned finally with the problem shall be given greater weight. From a representative sampling of their views, some aspects of which are presented below, we seem nearer in reality to the day when a cheap airplane will be made and marketed than at any time during the many years that the idea has been floated on the ocean of public consciousness, only to be brought up with a crash on the rock of technical achievement.

Questions to be Answered

As the problem stands, there are literally hundreds of questions concerning its details for which an answer is not available from the Department of Commerce, or from the engineers who have interested themselves in it. There exists also a large and qualified section of engineering opinion which regards the whole thing as a mirage which recurs during the comparatively dry seasons of the engineering art.

But another large and qualified section of the engineering world has implied that the project is worthy of discussion by the simple action of discussing it.

A meeting of the Philadelphia Section on Jan. 17 gave the first opportunity for the Society to hear officially about Mr. Vidal's project. Frederick R. Neely, chief, aeronautic information section of the Department of Commerce, described the Aeronautic Branch's plans and their progress. Members and guests of the Society and of The Aircrafters, who participated in the meeting, asked questions from every point of view of aircraft engineering.

The following statement of what the Aeronautics Branch of the Department of Commerce has set out to do is based on Mr. Neely's talk before the Philadelphia Section, and on supplementary information furnished by him for this article.

In the past, it was pointed out, the activities of the Aeronautics Branch has been associated, both in fact and in the public mind, with regulation of air-commerce details. There exists, however, in the Act creating the Branch a phrase giving the Director the duty of "fostering and promoting" the air commerce of the nation.

Establishment of the Public Works Administration made available funds for carrying out a coordinated plan for the development and extension of nationwide aviation activities. The possible development of a cheap airplane for wide distribution is one phase of the plan, which includes also:

(1) Use of PWA funds for the grading of approximately 2000 additional landing fields, which when completed will bring the total of landing fields of all types to 4000 for the United States.

(2) Increase in the mileage of lighted airways; particularly the equipment of the Northwest route to Puget Sound, and of the St. Louis to New Orleans Route.

(3) Approval and construction of the Armstrong Seadrome system for trans-oceanic flying.

The latter, Mr. Neely indicated, had been approved by the Department of Commerce and further progress awaited only the approval of the Department of State, which is investigating the international complications which might arise with the operation of such a system.

On the four points enumerated, including the light airplane development, the Aeronautics Branch bases its plans for the progress of American aviation. The landing-field and lighting features, being intensification of routine activities, have not received a great deal of public attention but are definitely in the action stage.

Armstrong Plans Held Up

The Armstrong system is temporarily shelved until all aspects of the problem are in the clear, and the project for the $700 airplane has reached a point where two committees, backed by 22 aircraft manufacturers, are cooperating with the Aeronautics Branch in working out technical, legal and business details connected with the plan. A third committee, appointed by Mr. Vidal, is reviewing other phases, including the aspects of presenting the idea to the public, and prospective buyers of the planes.

For carrying out the work of development of the $700 airplane project there is available $500,000 appropriated from PWA funds. The exact disposition of the money has not yet

DEPARTMENT OF COMMERCE
AERONAUTICS BRANCH
WASHINGTON

Memorandum to Department of Commerce Licensed Pilots, Student Pilots, and Mechanics

NOVEMBER 15, 1933.

We cannot hope for a natural, healthy, and widespread growth in the private flying business until we develop a product that will appeal to the man who can afford only a few hundred dollars for an airplane. Volume production and consumption of a popular-priced airplane will give the proper impetus to an increase in the number of airports, flying schools, flying clubs, and pilots who will fly only for pleasure.

It is a comparatively easy task to design and turn out on a volume-production scale a small airplane which will sell for around $700. Such a craft would be a 2-place, low-wing monoplane, constructed of a new steel alloy, fitted with an 8-cylinder small-bore engine, of about 4,000 r.p.m., and a geared propeller. The landing speed would be about 20 miles per hour, using air brakes.

The plane would be rugged and durable and one that would require very little attention. In fact, it is entirely possible for it to be operated for the full span of its life without major overhauls. Rather than subject the airplane to a major overhaul after several years of service, it probably would be more economical to purchase a new one when that time arrived. Arrangements would be made for selling the plane on a credit basis.

It occurs to us that many pilots, student pilots, and their friends would be interested in purchasing a plane of the general description and under the conditions mentioned above. However, before the aircraft industry undertakes volume production of such an airplane, it would need to know the extent of the market for such a product. If a favorable reaction follows, a first volume of 10,000 planes will be planned. Therefore, in an effort to assist the industry in this regard, and to lend all possible encouragement to the development, through volume production, of a popular priced, exceedingly safe and reliable airplane which will be easy to operate and economical to maintain, we are transmitting this communication for your reaction. Would you please fill out the questions given below, tear off, and return to us in the enclosed envelope, which requires no postage, at your earliest convenience.

(Signed) EUGENE L. VIDAL,
Director of Aeronautics.

(TEAR OFF HERE)

AERONAUTICS BRANCH,
DEPARTMENT OF COMMERCE,
Washington, D.C.

Q. Would you purchase a plane of the order described in the above memorandum if it could be made available in the near future?

A. .....

Q. If the answer is "No", please give your reasons

A. .....

Q. How many persons to your personal knowledge, in addition to yourself, would purchase such an airplane?

A. .....

Q. Do you know of any groups that would buy this type of airplane on a cooperative or community basis? If so, how many?

A. .....

Q. Do you own an airplane?

A. .....

Name .....

Address .....

Grade of license .....

(If more space is required for answers, please use either side or another sheet of paper)

Above is a condensed facsimile of the original Aeronautics Branch questionnaire initiating the light-airplane project

been determined and will depend to some extent on rulings by the Comptroller General as to whether it can be applied to certain specific problems as they arise.

A recent communication from the Department of Commerce states that 50,000 prospects for an airplane selling for about $700 have been collected. The work of collecting the names of sales prospects was the first step in the project as initiated by Mr. Vidal.

Questionnaires were mailed to 34,000 airmen (pilots, mechanics and students) on the Department of Commerce licensing list. The exact wording of the questionnaire is reproduced herewith.

Response to the questionnaire was rapid. In the first week after mailing 7661 replies were received; 5500 responders indicated that they would buy a plane of the type outlined, if it could be sold for about $700; 2000 responders said they could not or would not buy such a plane. One question asked was how many friends or acquaintances of the responder would be probable purchasers of the type of airplane discussed. The total number mentioned was 27,115.

Responses to the questionnaire continued to pour in. A complete analysis of the returns to Dec. 26 is shown below.

Report of Low-Priced Airplane Survey

	Definitely "Yes"	"Yes" with Comments	"Yes" Provisional	Total	Answered "No"	Total Returns
Pilots	3,357	1,678	1,695	6,730	2,805	9,535
Students	1,941	744	827	3,512	851	4,363
Mechanics	473	161	260	894	595	1,489
Others	950	435	659	2,044	891	2,935
	6,721	3,018	3,441	13,180	5,142	18,322

Answered "Yes" (Columns 1 and 2) ..... 9,739 or 53 per cent of returns
Answered "Yes" provisional.... 3,441 or 19 per cent of returns
Answered "No" ..... 5,142 or 28 per cent of returns

The following figures taken from the above returns indicate the NUMBER OF PERSONS (other than those answering the questionnaires) who to their personal knowledge would purchase an airplane of this type.

Reported by Pilots:	
Answering "Yes"	14,233
Answering "Yes" with comment	8,540
Answering "Yes" provisional	5,767
Answering "No"	5,348
Total	33,888
Reported by Students:	
Answering "Yes"	5,687
Answering "Yes" with comment	2,340
Answering "Yes" provisional	1,891
Answering "No"	1,003
Total	10,921
Reported by Mechanics:	
Answering "Yes"	2,150
Answering "Yes" with comment	621
Answering "Yes" provisional	729
Answering "No"	1,030
Total	4,530
Reported by Others:	
Answering "Yes"	3,819
Answering "Yes" with comment	1,875
Answering "Yes" provisional	1,232
Answering "No"	1,102
Total	8,028
Grand Total	57,367

Mr. Vidal has pointed out that there are in the United States 14,000 licensed pilots, 11,000 student pilots, 8500 licensed mechanics and only 7000 licensed aircraft.

"This condition exists", he has stated, "because the aircraft on the market today, owing to limited production, are high in original price, high in operating costs and require very

close attention with respect to maintenance. They are not within the reach of thousands of people whose automobiles must cost around seven or eight hundred dollars."

All that is really needed, Mr. Vidal believes, is some step that will bring the producer and consumer together, with definite assurance to the aircraft industry that a market for small planes actually exists. The Aeronautics Branch questionnaire was designed to supply the necessary link between the producer and consumer, and it was intended originally that when the results of the questionnaire were tabulated and the existence of a potential market established, the Aeronautics Branch would step out of the picture and let the industry itself determine what plans should be made for production.

With the appropriation of PWA funds for the project, however, and the appointment of a committee by Mr. Vidal to study further the possibilities of the project it is indicated that Department of Commerce participation may be extended beyond the limits first intended.

It has been suggested from many sides that production of

Three Designs Illustrating the Light Plane Art

Courtesy *Aviation*

Courtesy Aeronautical Chamber of Commerce of America

If the Department of Commerce project results in the production of a successful light airplane it will not be breaking entirely new ground.

The Taylor Cub (upper left), the Aeronca (upper right) are examples of two planes which have made available for private flying reliable ships of relatively low cost. The Curtiss-Wright Junior (below) is a pusher type which attracted a great deal of attention when it was introduced.

Courtesy *Aviation*

Two Aeronautical Chamber of Commerce Committees have been formed, the first with Ralph Damon (Curtiss-Wright, St. Louis) as chairman, to cooperate with the government in working out *technical details* and complete performance specifications of the new plane and engine. Clayton Brukner (Waco), B. D. DeWeese (Stinson), Major Fleet (Consolidated), and C. G. Taylor (Taylor Cub) make up the committee.

The second committee with Sherman Fairchild, chairman, is to work with the government in the business phases (legal and economic) of the practicability of a corporation to develop and produce the plane. Other members are: E. R. Breech (North American), P. G. Johnson (United), Thomas Morgan (Curtiss-Wright) and Taylor Stanley (Aeronca).

It is felt that the members of these committees, chosen because of their technical and organization experience, will coordinate the thought of the entire industry in its efforts to work out the $700 plane plan. Twenty-two manufacturers were present at the meeting in New York on January 8 when the committee members were chosen. These are as follows:

General Aviation Mfg. Corp.	Consolidated Aircraft Corp.
B/J Aircraft Corp.	Fleet Aircraft Co.
Fairchild Aviation Corp.	Amphibions, Inc.
Kreider-Reisner Aircraft Co.	Grover Loening Aircraft Co., Inc.
Waco Aircraft Co.	Curtiss-Wright Airplane Co.
Pitcairn Autogiro Co.	Curtiss Aeroplane & Motor Co., Inc.
Taylor Aircraft Co.	Stinson Aircraft Corp.
Don Luscombe	Aeronautical Corp. of America
Boeing Airplane Co.	Bellanca Aircraft Corp.
Sikorsky Aviation Corp.	
Chance Vought Corp.	
Stearman Aircraft Co.	
Kellett Autogiro Corp.	

10,000 light planes in a year could not be accomplished within the price estimate, unless the production were centered in one plant, under a single management. The whole aircraft industry, on the other hand, is supposed to benefit from the production and sales of the plane. It has been proposed, therefore, that a joint corporation be formed by interested aircraft manufacturers, possibly with government participation, and that this joint corporation should undertake the actual operations involved.

A committee headed by Sherman M. Fairchild is studying this phase of the problem for the Aeronautical Chamber of Commerce.

One of the points brought out by Mr. Neely was that an engine suitable for light planes had been developed in England to sell for the equivalent of $190. This pointed the way, he said, to what could be done if American manufacturers sought seriously to solve the problem of cost.

Discussers of the Philadelphia program noted, however, that the difference in labor costs alone would make the manufacture of such an engine in this country cost a great deal more than it would in England.

Mr. Neely offered other evidence to show what could be done to costs simply by raising present designs to a quantity-production basis. One airplane manufacturer approached by the Department of Commerce estimated that their product, which now sells for about $2500, could be sold for $650 if a production of 10,000 were attained within a year.

It was emphasized throughout both Mr. Neely's presentation and during the discussion that public acceptance of a cheap airplane would rest largely on whether the product fulfilled the public's conception of what an airplane should look like.

Monocoque or cantilever construction might have to be used, it was pointed out, because while a plane might be structurally sound with visible guys, etc., a public educated to streamlining would probably not respond enthusiastically to its appearance.

Reference to the questionnaire sent out by the Aeronautics Branch (reprinted herewith) will show all that has been stated about the type of plane which might be produced. Discussion at the Philadelphia meeting was necessarily limited to the problem as then stated.

The possible hazards of putting a large number of pilots into the air with relatively untried planes received immediate attention at the Philadelphia meeting. Is the project "a means of making flying possible for a large number of young men who might eventually form an irresponsible menace at the various airports of the country?", asked R. W. A. Brewer, consulting engineer with a long experience in aviation problems.

From the standpoint of production, Mr. Brewer said, he and his associates had spent considerable time and money on the possibilities of putting out a production of 10,000 planes, but were unable to get the cost figure down to anything like that suggested by the Aeronautics Branch.

Considerable modification in Department of Commerce requirements for airplanes would be necessary, he believes, before a plane could be produced at really low cost.

In the case of a light plane, he pointed out, the cost would tend to be in inverse ratio to the weight and that it was possible that low cost could not be reached in connection with low weight.

The questionnaire specified that the projected plane would be "economical to maintain". Several of those present at the Philadelphia meeting questioned whether this would be possible on any basis comparable to the operating cost of an automobile.

Favorable comment on this point was contributed by Joseph A. Simcock, vice-president, F. & S. Aircraft Corp., which operates a group of light planes for training and hire.

The average light plane consumes 3 gal. of gasoline and ¼ pt. of lubricating oil per hour at cruising speed, according to Mr. Simcock. "Based on existing prices at most airports", he said, "we find that our cost for fuel and oil is 80 cents per hr., or one cent per mile at cruising speed. Our own planes (Aeroncas) have flown from the Northeast Philadelphia Airport to New York (Newark Airport) and back for less than $1.50.

"Other costs, including maintenance, repairs, etc., average about 20 cents per hr., giving a total cost of approximately $1 per hr. Storage charges and other costs which vary widely with locality have been omitted purposely from the figures."

Mr. Simcock deprecated the possibility of a large number of light planes in the air constituting a public menace. Mistakes will be made, he said, but the chances for injury are much smaller in a light plane.

"During the past three years our light planes have flown more than 1000 hr. in student work", he continued. "Our records show that during this time we have replaced two propellers, two landing-gear struts and put on a few patches. The total cost, including labor, did not exceed $150."

If a plane could be produced for a figure near $700, Mr. Simcock pointed out, the large burden of depreciation cost on big planes which adds so much to the cost of flying instruction and private flying, would be largely eliminated.

(Continued on page 32)

Chronicle *and* Comment

"TO my mind, the engineer is the unsung hero of the automotive industry. He is victimized as it were by the shallowness of automobile advertising." In a confidential letter, so writes an engineer who many years ago reformed and entered the realms of merchandising and sales analysis. He goes on:

"Fleischmann's yeast and Lucky Strike cigarette organizations go to the ends of the earth to cook up more or less imaginary yet interesting things to talk about in their advertising. We, in the automobile industry, have the most interesting product that has ever been produced by man—but we seem to exercise nothing short of downright genius when it comes to keeping these interesting things a dark subject.

"C. F. Kettering has built up a wonderful and highly-deserved reputation on the foundation of the foregoing delinquency.

"Van Loon, DeKruif and the author of the 'Russian Primer' have risen to fame via the same route.

"The automotive engineers need an Ivy Lee, an E. L. Bernays—or a John the Baptist!"

OFF to its maiden voyage on the sea of national meeting planning, the recently organized Tractor and Industrial Power Equipment Committee already has sighted Port Success.

With almost every speaker definitely set and most details of the program completed in February for the first National Tractor and Industrial Power Equipment Meeting on April 18-19, this freshly active group sets an example of effective organization not always equalled by divisions of longer experience.

By
Norman G. Shidle

Milwaukee is to be the site of this April gathering. Headquarters will be at the Pfister Hotel. C. C. Krieger, chairman of Tractor Committee, is acting also as chairman of the Meetings Committee. His efforts, supported by the active cooperation of the Milwaukee and Chicago Sections, are responsible for the well-ordered arrangements. Sessions will be held on two mornings, two afternoons and one evening.

Both spark-ignition and Diesel engines will be discussed as related to tractors and industrial power equipment, as will agricultural-industrial connections in several of their phases. Relationship between the agricultural and industrial power industry and the petroleum industry will furnish another topic.

The location, the dates and the program all point to good attendance and a successful meeting.

TRUCK, Bus and Railcar Activity is now the official name of the group formerly designated as the Motorcoach and Motor Truck Activity.

The new name marks a natural fruition of the gradually increasing interest in railcar development which has taken place among Society members in recent years—and of the gradually increasing interest in automotive engineering that has been taking place among railcar engineers.

The change in name of this Activity, it is reasonable to expect, may be regarded as indicative of an avenue of interest which will be opened wider and wider as time goes on and as possibilities of cooperation between automotive and railcar interests grow, as they seem certain to do.

IN one of the most unusual statements we recall having emanated from so world-famous an industrialist, Sir Henri W. A. Deterding asks the motor-car industry of the United States if it has ever looked into the future and "realized that where Japan is able to produce bicycles that come on the market in Holland and the Dutch East Indies at U. S. $5 each, she will certainly be able to produce motor-cars at a great deal less than $500?"

"A Japanese motor-car industry," the internationally-known petroleum tycoon continues, "once it has been established with the aid of their cheap monetary unit, the Yen (. . . brought down as low as 20 cents against 50 cents as formerly)—being able itself to produce steel cheaply—and therefore only needing to buy rubber—is sure, in the long run, to produce cheap cars at about $100."

Other startling and hard-to-believe ideas expressed by Sir Henri include one that "China with her population of 500,000,000 of (very practical) inhabitants is in a position to use hundreds of thousands of motor cars yearly, provided that country is purged of communism and provided that there will be an end to the high financial 'grousing and grumbling' in order to bring down 'a tors et a travers' the monetary unit of that country to a low value by putting into circulation in Europe inferior shillings, florins and marks."

Behind the Scenes With

Tractor and Industrial Power Equipment

DURING the Annual Meeting of the Society, Chairman C. G. Krieger of the Agricultural and Industrial Tractor Committee that was organized by the Society following the International Automotive Engineering Congress in Chicago last August, worked with several members in the

agricultural tractor and industrial power equipment field, the program for a National Tractor and Industrial Power Equipment Meeting of the Society being planned by this committee.

This plan provides for a two-day meeting of technical sessions in Milwaukee on April 18 and 19 in which the members of the Chicago and Milwaukee Sections of the Society have shown a deep interest and desire to cooperate with the Society.

At the meetings of the National Meetings Committee and Council this program of the Tractor Committee was reviewed and approved so that immediate further progress toward final arrangements for the meeting can be completed.

C.F.R. Detonation

A TABULATION of the results obtained in the general reproducibility check tests by the C.F.R. Motor Method, (A.S.T.M. Tentative Method of Test for Knock Characteristics of Motor Fuels, Designation: D 357-33T) were reported to the Detonation Subcommittee at its Jan. 26 meeting and copies are being circulated to all participants.

The results of these tests substantiate those obtained in a series conducted within the Subcommittee itself and laboratories represented on the Cooperative Fuel Research Committee. A report covering the results of the earlier series of tests conducted by the Detonation Subcommittee was prepared and presented by H. F. Huf to Technical Committee A on gasoline of the American Society for Testing Materials at its meeting Jan. 20, and is released for publication.

Lubricants

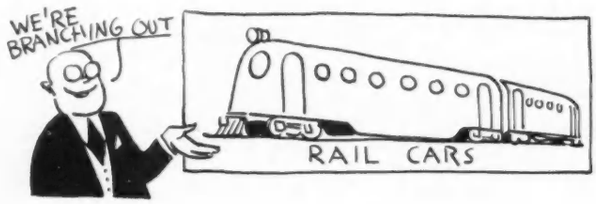
AN important meeting of the Standards Committee Lubricants Division was held Jan. 22. Progress was made toward the final draft of a set of lubricant data forms that it is proposed should be issued to car manufacturers for the collection of mechanical and lubricant specifications data for distribution by the Society, at the proper time, for general use. One of the principal purposes of this project is to save the car manufacturers both appreciable time and expense by enabling them to fill in only one master form for the Society in place of answering a large number of individual inquirers, as at present.

An important progress report on chassis lubricants was discussed by a subdivision during the morning and later by the Division, covering six classifications on this type of lubricants to meet various requirements in use. Another important step taken was to discontinue publishing general information on the free-wheeling lubricants classification as now published for trial in the S.A.E. HANDBOOK. A sub-division is to be appointed to study the present classification for transmission and rear-axle lubricants with particular reference to the classification of lubricant for Winter or low-temperature operation.

It was also decided that the Division members should cooperate with a special technical committee of the A.S.T.M. that is now studying the use of low-temperature crankcase oils with respect to their various properties and application, these studies relating to the dissemination of such information to car owners for the improvement of Winter operation.

Motorcoach and Motor-Truck

THE closing meeting of the 1933 committee was convened by Chairman M. C. Horine on Jan. 22. He reviewed the consideration that had been given to expanding the activity to include rail cars, and reported that the matter had been thoroughly discussed with all members



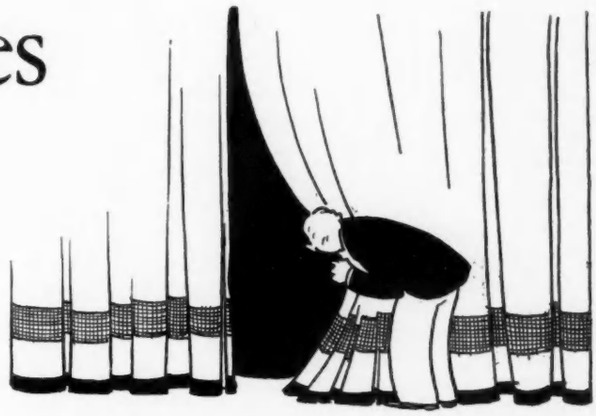
of the Activity committee and other members of the Society conversant with this field of transportation and that the Council had authorized this extension of the Activity's interests, changing the Activity's name to "the Truck, Bus and Railcar Activity."

L. R. Buckendale, chairman of the Activity Meetings Committee, reported on the sessions held at the meetings of the Society and also that during the year 16 sections had held 30 meetings at which a total of 35 M. & M. papers had been presented. He also stated that 14 papers had been published in 10 issues of the JOURNAL during 1933 totalling more than 50 pages of JOURNAL text.

The 1934 committee was then convened by Chairman A. K. Brumbaugh and a member of the Activity Nominating Committee was elected to select a vice-presidential nominee for 1935.

The Committee discussed in a general way an approach to its work in the railcar field and also further development of the Activity's programs among the sections of the Society.

the Committees



Aviation Gasoline Detonation

CHAIRMAN ARTHUR NUTT rendered a complete report of progress to date to his C.F.R. Aviation Gasoline Detonation Subcommittee at its meeting on Jan. 23.

Two of the five cooperating engine laboratories now have completed the initial portion of the test program.

The budgeting of the project for the coming year was considered and a specific plan for future financing was worked out and approved.

Production Activity

ON Jan. 25, F. W. Cederleaf, chairman of the Production Activity Committee convened the final meeting of the 1933 Committee and indicated in his review of the Committee's work that it was felt good progress had been made during the year.

Meetings Committee Chairman Geschelin then referred to the production sessions at the annual meetings of the Society and reported that 7 sections had prepared 9 production-topic papers that were discussed at 8 section meetings during the year and that 14 production papers consisting of more than 55 S.A.E. JOURNAL pages had been published in 9 issues of the JOURNAL during the year.

The 1934 Committee was convened by Chairman W. H. McCoy, who announced the personnel of the Committee and the chairmen of the subcommittees for the year. After electing a member of the Nominating Committee to select a vice-presidential nominee of the Activity for 1935, the Committee discussed at some length a possible program of production meetings during the year and the general topics of most importance in the production field. The Committee agreed that probably the two outstanding subjects suitable for meetings papers relate to the quality and methods of finishing metal surfaces in manufacturing and the employer-employee relationship under the new industrial order.

Standards

THERE was an exceptionally well-attended meeting of the Standards Committee Jan. 24, at the close of its year. Eight divisions submitted 15 reports, all of which were approved as submitted, except the non-ferrous metals, in which several modifications were made, subject to confirmation by the division reporting them.

These reports were submitted by Standards Committee Chairman Arthur Boor at the following meeting of the Council, at which the action taken by the Standards Committee was approved.

Accordingly, the several reports acted on will subsequently be issued to the members in the usual manner.

Research

IN addition to approving reports from the various Subcommittees, the main Research Committee at its meeting on Jan. 25 received summary reports from Dr. O. C. Bridgeman and H. K. Cummings covering progress on the fuels and lubricants projects at the Bureau of Standards.

Mr. Cummings reported that the investigation of aviation spark-plugs initiated last August under the joint sponsorship of the Society of Automotive Engineers and the Bureau of Aeronautics, Navy Department, is progressing satisfactorily. The extension of the investigation to types other than mica spark-plugs, and the development of some standard method of specifying the relative "coldness" of spark-plugs is under consideration, and it was therefore recommended that the Research Committee appoint a small Spark-Plug Subcommittee to keep in touch with the work on that project and serve in an advisory capacity.

The report and recommendation were approved and it is expected that Chairman R. R. Teetor will announce shortly the personnel of the new subcommittee.

Transportation and Maintenance

AT the final meeting of the 1933 Transportation and Maintenance Activity Committee, Vice-Chairman T. C. Smith, who presided in the absence abroad of Chairman J. F. Winchester, summarized in a general way the results of the Committee's and the Activity's work during the year. Chairman L. V. Newton of the Meetings Committee reported on the programs of the Activity at the annual meetings and that 23 sections had held 25 meetings at which 32 T & M papers had been presented. He also reported that during the year 17 T & M papers had been published in 10 issues of the JOURNAL, totalling more than 65 pages.

Mr. Newton, as Chairman of the Committee for 1934, took the chair and announced that F. K. Glynn and R. T. Hendrickson had been appointed chairmen of the Meetings and Membership Committees, respectively.

Fred L. Faulkner, Armour & Co., was named chairman of a subcommittee to study vehicle design from a maintenance and operating standpoint.

(*Continued on page* 24)

Design Changes Needed to Simplify

THE absolute necessity for designs giving adequate protection from dirt and water must be recognized as a predominant factor in any attempt to reduce the frequency of necessity for chassis lubrication, Mr. Churchill concludes in this striking discussion of a problem which concerns nearly everybody in the automotive industry directly or indirectly.

Inattention given to chassis lubrication problems, he says, is reflected in the increasing number of points to be lubricated and the necessity for frequent lubrication.

"Recommendations regarding proper lubricants and the frequency of their application," he states, "are made by the manufacturer on the basis of test experience which sometimes is quite meager.

"It is not uncommon to observe recommendations of highly specialized lubricants which have been developed to camouflage the inherent weakness of a poor mechanical design."

Backing his statements with a detailed analysis of data taken from lubrication recommendations of eleven representative car makers on models covering the period 1928 to 1933, Mr. Churchill writes frankly and specifically of existing troubles—and suggests the road along which progress lies.

PROBLEMS of chassis lubrication usually are considered of secondary importance to those involving performance and appearance. Inattention to these problems is often reflected in the increasing number of points to be lubricated and the necessity for frequent lubrication.

Recommendations regarding the proper lubricants and frequency of their application to the chassis elements are made by the manufacturer upon the basis of test experience which is sometimes quite meager. It is not uncommon to observe recommendations of highly specialized lubricants which are developed to camouflage the inherent weakness of a poor mechanical design. However, control and supervision of recommendations is remote and very often entirely lacking.

It is necessary in the design of chassis elements to consider the factor of frequency of lubrication and to reduce the number of points lubricated if the hazard to reliability, over which control is limited, is to be minimized and if the present chaotic situation regarding the multitude of special oils and variance of opinions as to requirements is to be simplified.

An analysis has been made of the changes in number of points to be lubricated and their relative frequencies of lubrication. The data has been taken from the recommendations of eleven representative manufacturers. This survey covers the period from 1928 to 1933, and the data used were taken from the recommendations pertaining to the successors of the original 1928 model of each respective manufacturer.

The change in total number of points requiring lubrication (Table 1) shows that a low was established in 1929 and a high reached in the 1932 models. The low of 46 in 1929 was due to the adoption of rubber shackles requiring no lubrication on two of the models considered. The high of 55 established in 1932 was due to the general introduction of powerplant accessories requiring lubrication and the adoption of ride control on several of the makes contained in the survey.

The number of lubricants specified (Table 2) has shown a gradual increase through the period of this analysis. Inclusion of the various manufacturer's specifications of lubricants for winter and summer operation would almost double the increase indicated. Development of more suitable lubricants for specific types of duty accounts for the gradual increase. This necessarily complicates the problem of the private owner complying with manufacturer's recommendations.

A more thorough study of design for the exclusion of dirt and water and improvements in load-carrying ability of chassis-element bearings would serve to reduce materially the number of lubricants required and to simplify their specifications. A comparison of the frequency of lubrication of the chassis, powerplant and drive-line elements of the car (Table

Table 1—Change in Total Number of Points Requiring Lubrication 1928-1933

Average of 11 representative makes of cars used as basis for these data

Year	Chassis	Powerplant	Drive Line	Total No.
1928	36.4	7.1	4.3	47.8
1929	34.6	7.3	4.2	46.1
1930	39.2	8.2	4.6	52.0
1931	37.0	9.2	4.6	50.8
1932	39.2	11.0	4.9	55.1
1933	38.8	9.0	4.5	52.3

Table 2—Change in Total Number of Lubricants

Recommendations of various specifications for winter and summer were not tabulated

Year	Chassis	Powerplant	Drive Line	Total
1928	2.1	3.1	2.4	4.0
1929	2.0	2.8	2.5	4.0
1930	2.5	3.6	2.7	4.4
1931	2.8	3.8	2.8	5.3
1932	2.7	3.9	2.6	5.3
1933	3.2	3.9	3.0	5.7

[This paper was presented at the Annual Meeting of the Society, Detroit, January, 1934.]

Chassis Lubrication Problems

By H. E. Churchill
Research Dept., Studebaker Corp. of America

3) shows that simplification is being effected on chassis elements as the number of points requiring infrequent attention have increased from 1928 to 1933. The number of points requiring infrequent lubrication has increased 140 per cent since 1928 whereas there has only been an increase of approximately 10 per cent in the total. This is attributable to the increased use of non-friction bearings, provided with effective seals for retention of the lubricant and the exclusion of dirt, even though costs are slightly increased.

The frequency ratios of powerplant and drive-line lubrication has undergone relatively little change but the number of points has shown a slight increase principally because accessories are continually being added. The duty on these elements has also gradually become more severe due to increases in car and engine speeds. Frequent lubrication of powerplant

Table 3—Comparison of Frequency of Lubrication of Chassis, Powerplant and Drive-Line Elements

	Points of Frequent Lubrication			Points of Infrequent Lubrication		
Year	Chassis	Power-plant	Drive Line	Chassis	Power-plant	Drive Line
1928	31.3	7.1	2.0	5.1	0	2.3
1929	31.5	6.8	2.0	5.0	.5	2.2
1930	31.0	8.0	2.3	8.2	.3	2.3
1931	27.8	9.0	2.2	9.2	.3	2.6
1932	27.0	9.7	1.8	12.2	.5	3.1
1933	26.8	8.5	1.5	12.0	.5	3.0

Table 4—Comparative Total Number of Points Requiring Lubrication on Cars in Low, Medium and High-Priced Classes

	Total Points		
Year	Low Price Class	Medium Price Class	High Price Class
1928	56.0	42.3	55.0
1929	39.2	43.4	55.9
1930	52.2	46.6	60.9
1931	51.6	40.8	66.3
1932	51.3	46.0	63.3
1933	50.6	45.6	63.5

elements does not seriously complicate our problems because of their accessibility.

Practically universal adoption of needle bearings in the trunnions of propeller-shaft joints is responsible for the change indicated in the drive-line elements.

The recommendations for the 11 cars have been subdivided, on the basis of price classes to show the relative frequency and number of points lubricated. These average data (Tables 4 and 5) were obtained from 3 cars each in the high and low-priced field and from 5 cars in the medium-priced field.

Both the high and low-priced car averages show a decided increase in the number of frequent and infrequent points of lubrication; whereas, the medium-priced class has remained relatively stable as regards total number of points to be lubricated. However, the medium-priced class average shows a marked reduction in the average number of frequent points of lubrication, with a very gratifying increase over the six-year period of 220 per cent in the number of points of infrequent lubrication.

Necessity for economical design in the low-priced class prohibits, in many cases, consideration of cost increases necessary to decrease the total number of points or the frequency of lubrication. This is not the case in the medium and high-priced brackets.

The high-priced cars are usually chauffeur-driven and are serviced in accordance with manufacturer's recommendations. They are equipped with numerous accessories and, because of their usual large size and increased loading of chassis elements, have been provided with the necessary points for adequate lubrication without undue apprehension on the part of the manufacturer regarding compliance with lubricating recommendations.

Exploitation of chassis features in advertising and as a part of sales psychology has concentrated the greatest change, as regards lubrication, in the medium-priced field. This has resulted in development of numerous applications of non-lubricated bearings and a decrease in frequency of lubrication through the use of "non-friction" bearings with more adequate protection.

As previously shown in Table 3, the greatest transition from frequent to infrequent lubrication has been in the chassis rather than the powerplant or drive-line elements of the car. A further price-class comparison in Tables 6 and 7 of chassis elements only reveals that the greatest improvement or development has been in the medium-priced cars. This improvement has generally effected the design of brake linkages, shackles, shock-absorber links, steering linkages and propeller shafts.

Brake linkages have undergone greater development than

any other single element on the chassis if development is measured in the relative reduction of points requiring lubricant and the relative change in frequency of lubrication. The brake linkage system of a certain 1926 model car had 23 points requiring frequent lubrication, while the 1934 model had 4 points requiring lubrication or inspection every 10,000 miles and 7 points requiring attention every 2500 miles.

The rod link and lever linkage has been antiquated by cable and conduit control or hydraulic braking systems. The joints or bearings of the former were exposed to dirt, dust and water and were never free from rattles. Oilless bearings in the majority of applications were unsuccessfully tried on several models from 1926 to 1930. This design was more costly to use and brought no more satisfactory results because, in most instances, protection from dirt and water was sadly lacking. Fortunately, this experience did not condemn oilless bushings nor retard their future development, but did show the necessity for protection equally as good as that given anti-friction bearings. Oilless bushing installations provided with adequate protection are now functioning satisfactorily on many chassis elements.

THE total number of points requiring lubrication was lowest in 1929 and highest in 1932.

The low point was due to use of rubber shackles requiring no lubrication on two of the 11 models considered.

The 1932 high resulted from general introduction of powerplant accessories requiring lubrication and adoption of ride-control by several companies.

Cable and Conduit Type Control

The cable and conduit type of control made its debut in 1929 and has since come into general use on all price classes of cars. The usual construction providing very adequate protection from dirt and water requires only one point of infrequent lubrication for each cable and one of frequent lubrication where it attaches to the cross-shaft.

Whether or not clevis and pin joints, in link or clevis constructions exposed to dirt, should be lubricated is a subject in which arguments pro and con are presented. It is the writer's opinion that the detrimental effect of grit suspended in or attracted to the bearing by the presence of oil or grease is more detrimental as regards lapping and wear than the slight increase in friction and possibility of rattles if the joint received no lubricant. The possibility of noise is remote if tension in the linkage is provided by adequate pedal retracting springs.

Necessity for protection of ball or roller bearings from foreign matter has long been recognized and the resultant protection applied to wheel bearings, both front and rear, enables them to function properly with infrequent inspection. This is equally true on the three predominant types of rear-wheel bearing construction. The type of lubricant recommended must have saponifiable materials sufficient to keep it in the bearing and prevent it from being thrown out on braking surfaces which become erratic in the presence of grease.

Steering knuckle bearings are subject to severe shock and static loading under conditions of slow oscillating motion. The predominant design of today incorporates plain bearings oscillating on hardened pins. These bushings, due to the type of loading and motion, are subject to seizure unless provision is made for adequate lubrication.

Test results have proven that, for given loadings, grease-lubricated knuckles are more subject to seizure than oil-lubricated knuckles in duplicate service. Oscillating motion of this type has a tendency to wipe lubricant from the bearing surface and a lubricant with high film strength and great permeability is desirable. The use of chromium-plated knuckle pins with extremely hard surfaces are not as susceptible to seizure as plain carburized and hardened pins.

Grease lubrication of knuckle pins necessitates the provision of a greasing attachment for each bearing if proper distribution is obtained because of its comparative lack of permeability and the fact that the path of least resistance seems to be toward the unloaded portion of the bearing.

Oil lubrication of knuckle pins permits reduction of the number of points lubricated and the frequency of lubrication through the provision of magazine oilers and wicking systems for distribution. Distribution can be obtained and maintained by proper design.

Front axle designs incorporating needle bearings at the knuckles have been subjected to rigorous tests and have performed satisfactorily. Some of these designs have undergone road tests of 30,000 miles without any indication of failure or loss of lubricant. The adoption of a design of this type would improve steering qualities, particularly on heavy cars, and eliminate the present necessity of lubrication and frequent rebushing of steering knuckles.

The basic design of steering linkages has remained unchanged until the recent introduction of independent suspensions. Lubrication of the linkage joints is required principally because of inadequate seals for their protection from dirt and water. There have been several developments of interest which have tended toward reduction of frequency or elimination of lubrication.

Table 5—Comparative Number of Points Requiring Frequent and Infrequent Lubrication

	No. of Frequent Points of Lubrication			No. of Infrequent Points of Lubrication		
Year	Low Price Class	Medium Price Class	High Price Class	Low Price Class	Medium Price Class	High Price Class
1928		37	45.5		5.3	9.5
1929	30.6	37	47.3	8.9	6.4	8.6
1930	39.6	37	50.3	12.6	9.6	10.6
1931	39.0	29.4	54.0	12.6	11.4	12.3
1932	38.0	32.2	49.0	13.3	13.8	14.3
1933	37.3	28.8	49.0	13.3	16.8	14.6

Table 6—Comparative Total Number of Chassis Points Requiring Lubrication on Different-Priced Classes of Cars

	Price Class		
Year	Low	Medium	High
1928	——	30.8	44.5
1929	29.9	31.0	44.6
1930	40.9	33.0	47.6
1931	40.9	26.4	50.3
1932	36.9	34.4	48.9
1933	37.0	33.6	48.9

Table 7—Comparative Number of Chassis Points Requiring Frequent and Infrequent Lubrication on Different-Priced Classes of Cars

	Points of Frequent Lubrication			Points of Infrequent Lubrication		
Year	Low Price	Medium Price	High Price	Low Price	Medium Price	High Price
1928		27.5	37.5		3.3	7.0
1929	24.3	27.0	38.3	5.6	4.0	6.3
1930	31.3	25.8	39.3	9.6	7.2	8.3
1931	31.3	18.0	40.3	9.6	8.4	10.0
1932	27.3	20.2	37.6	9.6	14.2	11.3
1933	27.0	20.2	37.6	10.0	13.4	11.3

Those joints requiring no lubrication incorporate a combination of oilless bushings and rubber, or rubber alone bonded to the ball stud and housing. The angularity through which reach-rod joints must operate, due to axle roll, axle jounce and turning, prevents their general adoption at this point.

Wick lubrication of the reach-rod joints has been successfully used with a very marked reduction in frequency of lubrication. The usual tubular construction provides an excellent magazine for oil which is wicked through the magazine plugs and ball sockets directly to the working surfaces of the joint. A constant replenishment of oil in the bearing serves to keep out dirt which otherwise would obtain entrance through the usual inadequate dirt seal.

Tests of Ball Bearing Joints

Ball bearing joints of two different types have been subjected to tests with satisfactory results but have not yet been used as production equipment. These designs are packed with lubricant and thereafter require infrequent inspection. This design reduces friction in the steering linkage which produces a very marked reduction in steering effort. The reduction of friction obtained in the linkage by the use of anti-friction bearings in connecting links and steering knuckles in some instances has required an increased friction in the knuckle thrust bearings, or at other points, to eliminate or reduce the susceptibility of the car to shimmy and low speed wobble.

The steering gear of 1920 usually consisted of a pair of bevel gears installed under the hood and exposed to dust. They were susceptible to rapid wear as soon as the protective medium of heavy grease was lost. Realization of the fallacy of grease as a protection from dust and dirt resulted in the design of a very effective seal in the form of protective encasement or housing. Further development of this housing, as regards leakage, today permits the use of fluid lubricants.

Increased efficiencies obtained by liberal use of anti-friction bearings has produced a remarkable ease of steering and freedom from rattles. The refined designs of today, due to careful study and consideration of problems involved, are satisfactorily lubricated in winter and summer with lubricants of the same specification.

Until the introduction of two-way shock absorbers, the connection to the axle consisted of straps requiring no lubrication even though they were boldly exposed to dust, mud and water.

The linkage of hydraulic two-way shock absorbers must, of necessity, be a rigid connection incorporating two spherical bearings subject to oscillatory motion. The linkages, due to their bold exposure, demanded frequent flushing and lubrication, principally because of inadequate seals. The relative inaccessibility of these points of lubrication has resulted in general adoption of oilless bushing materials or rubber in the link bearings. Their development has also been such that sufficiently adequate protection has been generally incorporated.

Spring shackles may be subdivided into three classes; those requiring no lubrication, those requiring frequent lubrication, and those requiring infrequent lubrication. The fundamental difference in those requiring frequent and infrequent lubrication is in the adequacy of protection from dirt and water.

Non-Lubricated Shackles

Non-lubricated shackles are composed of rubber or other flexible elements. The loads transmitted and their resultant angular motion produce molecular flow without relative motion, thus eliminating the necessity for lubrication. However, this type of shackle soon deteriorates if brought in contact with abrasive dirt at the more highly stressed parts of the bearing. Fortunately, these more highly stressed parts are very well protected by material which has little to do except prevent the entrance of abrasive materials.

Installations of metallic shackles requiring frequent lubrication have received little consideration regarding protection from dirt and water. Lubricant frequently is applied in the hope that it will serve as a seal.

THE number of lubricants specified has increased gradually from 1928 to 1933, due to development of more suitable lubricants for specific types of duty.

Number of chassis points requiring infrequent lubrication has increased 140 per cent in this same period, although the total number of lubrication points increased only 10 per cent.

Slight increase has taken place in number of lubrication points of powerplant and drive-line elements, but the lubrication frequency ratio in these elements has undergone little change since 1928.

The bolt-and-bushing type of shackle was sorely lacking in bearing protection and subject to excessive wear, rattles and seizure because of the entrance of dirt. It has been proven that oil is the better type of lubricant for the oscillating motion and uni-directional loading of plain shackle bearings, but lubricating qualities were necessarily sacrificed to grease which served the dual purpose of seal and lubricant.

The screw type of shackle is inherently better protected than the plain type because the outer threads of the bearing act as a baffle for dirt and water. Foreign matter must travel an interrupted helical course to become deposited in the inner bearing surfaces. In some instances additional protection has been provided in the form of felt seals at the outer ends of the bearing. Tests have shown the additional protection noticeably prolongs the interval of lubrication.

Ball-bearing shackles are subjected to the same loading and angular motion as those of any other type. However, they

do not require frequent attention because it was demonstrated early in their development that an adequate seal to exclude dirt and water was necessary. The effectiveness of the seal used is demonstrated by the fact that some installations have run 50,000 miles with the original filling of grease and without any contamination of the bearing by dirt and water. The success of this design can be attributed primarily to the consideration which has been given the problem of protection.

Frequent lubrication of propeller-shaft joints has been necessary because of loss of lubricant through inadequate grease seals. Special fibre greases were specified on some designs because of their tenacity and resistance to centrifugal force. Recent developments indicate the consideration given this problem with the resultant abandonment of a seal which provides for the spherical motion required to seal the entire joint. It is general practice today to provide a seal for each individual trunnion bearing which only requires oscillatory motion. The problem of joint lubrication has been further simplified by adoption of needle bearing units on the trunnions with only rolling friction in place of the plain type of pin and bushing with sliding friction.

The foregoing developments have eliminated the necessity for frequent lubrication and some manufacturers recommend that the propeller-shaft joints be given no attention whatsoever.

Frequent lubrication of propeller-shaft splines would be unnecessary if a suitable seal were developed for the retention of lubricant.

This problem seems to be nearing solution by the removal of the spline from the shaft. Incorporation of the spline on the rear axle pinion or the transmission main shaft with lubrication therefrom will eliminate the necessity for propeller-shaft lubrication.

Future Complexities

Manufacturers and service organizations are confronted with a further complication of the problem of chassis lubrication if the present relative rate of change is continued. The trend toward streamlining of bodies will make the chassis elements requiring lubrication more inaccessible and therefore more susceptible to neglect. It is hardly believable that the use of streamlined bodies will reduce the amount or character of road dirt which is the basic source of trouble the chassis lubrication engineer has to combat. Independent wheel suspensions by their very nature, must necessarily add to the already large number of bearings or joints on the chassis which are subjected to static loading and slow speed oscillatory motion.

The ideal car, at least from the owner's standpoint, would be one which required no attention. However, the present objective should be a compromise in which the problem would be materially simplified by a uniformly scheduled annual or semi-annual inspection and replenishment of the lubricant in each of the several chassis elements. Designs suited to such a program would be of such a nature that simplification of lubricants available and specifications of the same would automatically follow.

The present complication can be relieved only by following a uniform program of development suggested by the foregoing discussion, in which the absolute necessity for designs giving adequate protection from dirt and water is recognized as the predominant factor determining the frequency of chassis lubrication.

Changes in By-Laws

In accordance with the requirements of the Constitution, the following amendments to the by-laws, approved by the Council at its meeting on Jan. 23, 1934, are published herewith:

In B39, change "Motor-Truck and Motorcoach" to read "Truck, Bus and Railcar".

Add the following paragraph to B31: "The books of the Society are to be closed each year on September 30, and any balance from Section dues or appropriations not yet paid to the Sections, and not needed for starting their fall work shall be transferred to the general funds of the Society."

Behind the Scenes With the Committees

(*Continued from page* 19)

J. M. Orr, Equitable Auto Co., is to be chairman of a subcommittee to study accident control in fleet operation.

After electing a member of the T & M Nominating Committee that is to select a candidate for the T & M vice-presidency of the Society for 1935, and discussing briefly the general program of the Activity committee for this year, the meeting adjourned.

E. L. Tirrell was appointed chairman of a subcommittee to study weight distribution on the front and rear axles of motor-trucks, including the CA dimensions (for which there is an S.A.E. Standard, p. 98, 1933 S. A. E. Handbook) and also the location of tractor-trailer fifth-wheels. The study relates to trucks of ½ ton nominal capacity and larger.

Motor-Truck Rating

On Jan. 24, Chairman L. R. Buckendale convened the Motor-Truck Rating Committee which reviewed its activities and deliberations to date and the results of a recent survey as to the use of the suggested method of rating of motor-trucks that had been released by the Committee some time ago. This survey indicated that the suggested method is considered generally too complicated for use and as a result of discussion a much simpler proposal was drafted that will be considered further by the Committee and probably circularized later.

Riding Comfort Research

At a meeting of the Riding Comfort Research Subcommittee on Jan. 24 the matter of future manufacture of the Wabblemeter was discussed. Four of the 19 claims in the patent application on the instrument have been allowed, it was stated, and possible extensive commercial applications of the machine will be investigated. F. F. Chandler, Ross Gear & Tool Co., was given authority to negotiate for manufacture and sale. The Passenger-Car Activity Committee will be urged to include a paper on the Summer Meeting schedule summarizing all work to date on the question of riding-comfort research.

Independent Wheel Suspension—Its Whys and Wherefores

By Maurice Olley

Special Problems Engineer, Cadillac Motor Car Co.

"ALTHOUGH it is not presented as a narrative," Mr. Olley says, "this paper is really the story of our work for the past three years. The sequence of the paper follows almost exactly the development of the story."

Independent suspension affects two things, it is pointed out—ride and steering. The first part of the paper discusses ride; the second part, independent suspension as it affects both ride and steering.

Mr. Olley offers no apology for injecting ride into the discussion, because, he says, "this is the reason why front suspension has been changed". He adds: "It has always seemed to me that the most interesting thing about any design is not the design itself, but the reason behind it." This paper gives the reasons.

IF a criticism may be offered of some of the work which has been done toward the improvement of ride, it is that the approach has been too indirect; too much along the lines of instrumentation rather than a frontal attack on the problem to find what has made cars ride as they did, and to improve them.

A common objection to such direct attack has been that riding quality was an "intangible," not directly measurable, and even a matter of personal preference and opinion.

The same objections might be applied to the factor of appearance, yet few will deny that the appearance of cars has been greatly improved in recent years.

In the same way we believed that it might be possible to construct automobiles in which the ride was so radically improved that there could be no two opinions about it, and no need for fine instrumentation to detect the difference.

This belief was based on experiments with small spring-suspended models, which were later confirmed on the road, as described later.

[This paper was presented at the Annual Meeting of the Society, Detroit, January, 1934.]

Two forms of instrumentation have proved extremely useful—the "bouncing table" and any simple form of instrument which will show the component frequencies of the ride.

The bouncing table (Fig. 1), as developed at General Motors Proving Ground, is a table or seat oscillating vertically with simple harmonic motion, and having adjustable stroke and frequency.

The "victim" is seated on this device, the stroke is set at any desired figure and the frequency is increased until the passenger says that he begins to feel uncomfortable.

By repeating the test at several different stroke settings and on a large number of subjects a "curve of comfort" is obtained which is fundamental. (Fig. 2.)

The range of variation between individual tests is small enough to show that essentially, riding comfort is not a matter of opinion.

In a motor car in which amplitudes of the order of 4 in. are to be expected, a frequency of 80 cycles per min. appears to be close enough to discomfort to be set as a practical maximum.

Human tolerance for frequencies of the order of 250 cycles per min. and up is so small that these "shakes" obviously must be suppressed as far as possible.

Frequently analysis on a car on the road shows that the average road at normal speeds disturbs the passage of the car by acting as excitation for the natural frequencies of the car itself. The motion of the car in other words is never a true picture of the road surface but is made up of the car-frequencies excited by that particular road surface.

The principal car oscillations which are in question are:

(1) The main frequencies—ranging from 60 to 150 cycles per min. in different cars.

(2) The tire frequencies—250 to 350 cycles per min. due to the car oscillating on its tires and without movement of the springs. Especially noticeable if there is excessive friction in the suspension.

(3) The frequencies of the unsprung masses—400 to 600 cycles per min. depending on the type of tires used and on the unsprung mass. The peculiarity of this group is that it has a tendency to start at the high frequency and slow down as the amplitude increases, due to the tires leaving the ground (like the bouncing of a rubber ball).

The first group is partly damped by the shock absorbers

and the friction of the suspension but contains the element of tire flexibility which is practically undamped. If the damping or friction (but particularly the friction) is too high, the second group enters the picture and becomes objectionable.

The third group can be fully damped provided the frame itself and the shock absorbers and connections are rigid enough and have a high enough natural frequency to escape resonance with the unsprung oscillations.

To hold these unsprung frequencies within reasonable limits generally requires a certain amount of friction or else a damping characteristic which simulates dry friction.

There is, therefore, a contradiction in the requirements for comfort between the second and third group, which requires a compromise both as regards friction and damping.

It becomes more difficult to find an acceptable compromise as the unsprung masses are increased.

Different types of road excite frequencies of the three groups to different extents.

Speaking very generally: Poor concrete and macadam chiefly excites—Group 1; good concrete—Group 2; rough roads, "washboard," etc.—Group 3.

We often read that suspension problems should be unknown in the United States because our roads are so good.

Actually this seems to be a complete fallacy. As compared with European roads, American highways abound in long waves, which are probably the result of extreme heat and cold.

We know both from experience and calculation that to hold a car at speed on such surfaces requires two qualities in the suspension—first a soft suspension, secondly a form of damping which will offer considerable resistance to movements of small velocity. Undoubtedly this accounts for the preference for softer suspension in America than elsewhere.

In springing a car we are not directly concerned with the rates of the springs but only with their deflection. This deflection is nominally the distance through which the spring is compressed from the unloaded to the loaded position. Actually measuring the deflection directly in this way gives poor results because in practically every suspension the rate varies throughout the range. What we are really concerned with therefore is the ratio of normal sprung load to the rate of the spring at the point of normal load. So that we are forced to get our "deflection" figures from measurements of the spring rate.

In a leaf-spring suspension the usual measurements of spring rate on individual springs with the center of the spring unclamped, is only the roughest sort of indication of the actual rate of the car suspension.

Even if a clamp representing the axle pad is bolted to the spring before testing, the result does not generally agree with the actual suspension rate, because of effects of shackle angle, etc.

The only test of spring rate which we have found even approximately reliable is to mount the complete car on four platform scales and pull it down to normal load position with the wheels resting on the scales. This is done by special jacks bolted to T slots in the floor, the jacks being arranged both to push and pull. (Fig. 3)

By then pulling the car down and pushing it back up again through a range of 3 or 4 in. either side of the normal load position and measuring deflections from the scales to a mark on each fender we can obtain a true load-deflection diagram for the suspension as a whole including the tires. At the same time we get a loop representing static friction in the suspension. The tire effect can easily be eliminated if desired by simple calculation knowing the tire-rate at normal load. Also by lifting one side of the car and lowering the other we can get the actual rate of the suspension in "rolling," which has supplied useful information on cornering stability.

Leaf-spring suspensions tested so far on all types of cars have shown stiffening effects of 10 per cent to 35 per cent

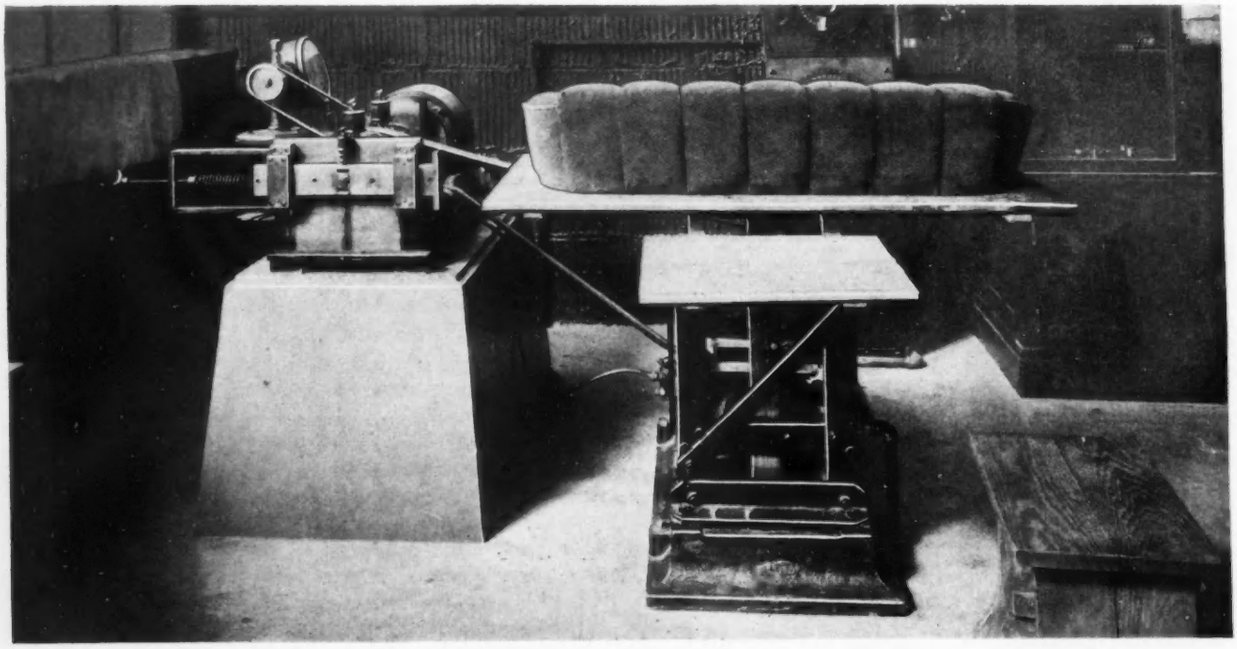


Fig. 1—The Bouncing Table, developed at General Motors Proving Ground, is a seat oscillating vertically with simple harmonic motion, having adjustable stroke and frequency.

above the rate of the unclamped spring in "pure ride," that is straight up and down movement. In pure roll, the spring rate in leaf spring suspension is about 50 per cent higher than in ride due to twisting of the springs.

If we accept the figure of 80 cycles per min. as a reasonable maximum for comfort we can use the simple basic formula:—

$$d = \left(\frac{188}{\text{Frequency per minute}}\right)^2$$

where d is deflection in inches, to determine at least what

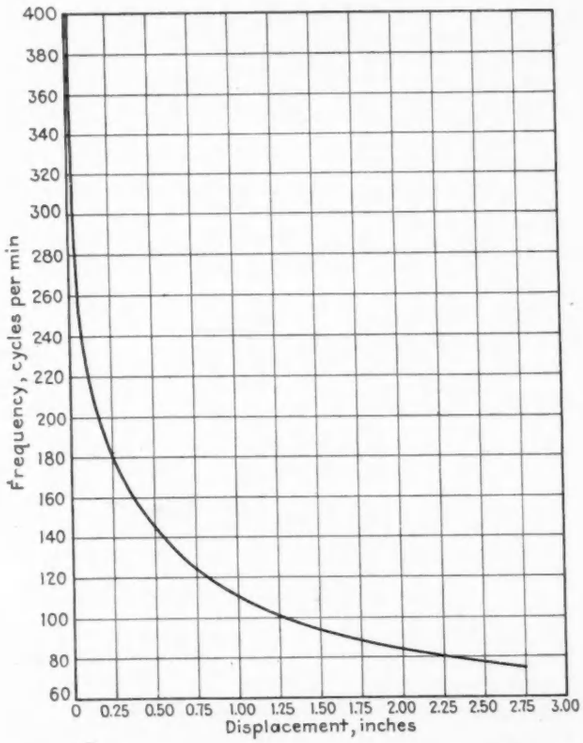


Fig. 2—"Curve of Comfort" Derived from Recording Results of Tests Made on the Bouncing Table

range of spring deflection to avoid. In this case the basic deflection is

$$\left(\frac{188}{80}\right)^2 = 5\tfrac{1}{2} \text{ in.}$$

So that we have established that the least spring deflection we ought to consider on a motor car is 5½ in.

Motions of Car

The main motions of a car conventionally sprung are: *pitch* —which is a rotary oscillation about a center within the wheelbase—and *bounce*—which is a similar but slower oscillation about a center outside the wheelbase.

The pitch frequency is 50 to 60 per cent faster than bounce. The pitching is chiefly oscillation of the front end but is about 15 per cent faster than the basic frequency of the front end. The bouncing is chiefly oscillation of the rear end but is about 10 per cent faster than the basic frequency of the rear end.

For example, suppose front and rear deflections of 4 in.

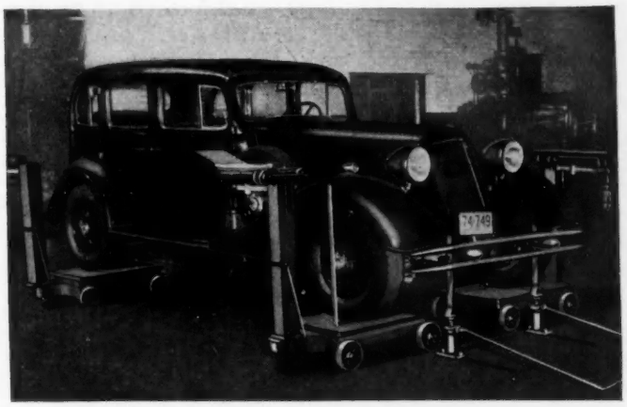


Fig. 3—The "Spring Test Rig" Used for Testing Spring Rates

and 9 in. respectively, basic frequencies will be 94 and 63 cycles per min. but actual frequencies will be about 110 and 70 cycles per min. So the conventional car does not achieve as low a frequency on either end as the same spring deflections would give in a simple system.

Recalling the great sensitiveness to frequency shown by the "comfort curve" this is the first disadvantage of conventional springing, and we can say that our earlier figure of 5½ in. minimum deflection should certainly be made 6 or 6½ in. to get a really comfortable ride.

The positions of the two centers are important, quite apart from frequency, because they determine the sort of "action" which the car will have (speaking of this as equivalent to the "action" of a horse). On the conventional car the bounce center is close to the front bumper and the pitch center near the back of the front seat. These centers are fixed, not by the actual deflections of the front and rear ends, but by the *relative* deflection of the two ends and the polar-moment of the car. So we can reproduce the same type of action with either stiff springs or soft springs, and can divide up the consideration of ride into two separate sections, one concerned with the action and the other with the "frequency level."

Action of Oscillation Centers

Some of the characteristics of these centers are:

(a) Whatever the disturbance of the car as a whole the motion of each center is simple harmonic.

(b) Adding load at either center does not affect the frequency about that center but slows down the frequency about the other. (For example on a conventional car removing the front bumper will speed up the pitch but will not affect the bounce).

To produce pure motion of either sort, the car must be struck at the opposite center, i. e., at the bounce center to produce pitch and at the pitch center to produce bounce.

In finding the frequencies of either end of a car it is usual either to stand on the bumpers and jerk the car into action, or to "pump" the bumpers by hand. The first method is obviously wrong as the weight of the man slows down the action. The second method will give a good indication of pitch frequency, but at the rear will not give the true bounce frequency because the rear bumper is, say 40 in. back of the rear wheels or 70 in. away from the pitch center. Pumping at the rear bumper causes a forced mixed vibration higher—and in some cases considerably higher—

than the true bounce frequency. The running boards just forward of the rear fenders, or even the rear doors, swung open, are better for getting true bounce frequency.

On the road the car gets its impulses through the wheels, and in the conventional car these are close enough to the centers so that the front blows chiefly cause pitch, and rear blows bounce; so as the car is driven along the road the faster motion is started first. (Fig. 4.) The front end is

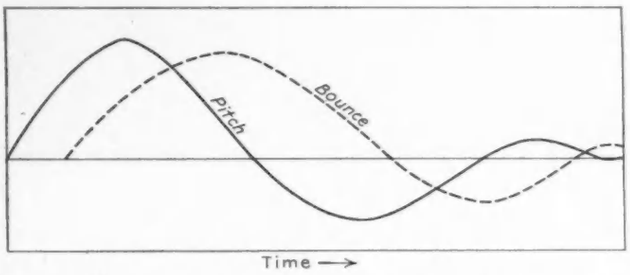


Fig. 4—Pitch and Bounce Sequence on Car Conventionally Sprung

dropping as the rear is rising and vice versa, so that a maximum amount of angular motion is produced by a single bump passing from front to rear. Here, therefore, is objection No. 2 to conventional springing.

Now consider that we start softening the front springs. A characteristic of the two centers is that softening front springs relative to rear makes both centers move forward and vice versa. The way in which these centers move when the suspension at one end is continuously softened and at the other end continuously stiffened, is shown in Fig. 5. As the front

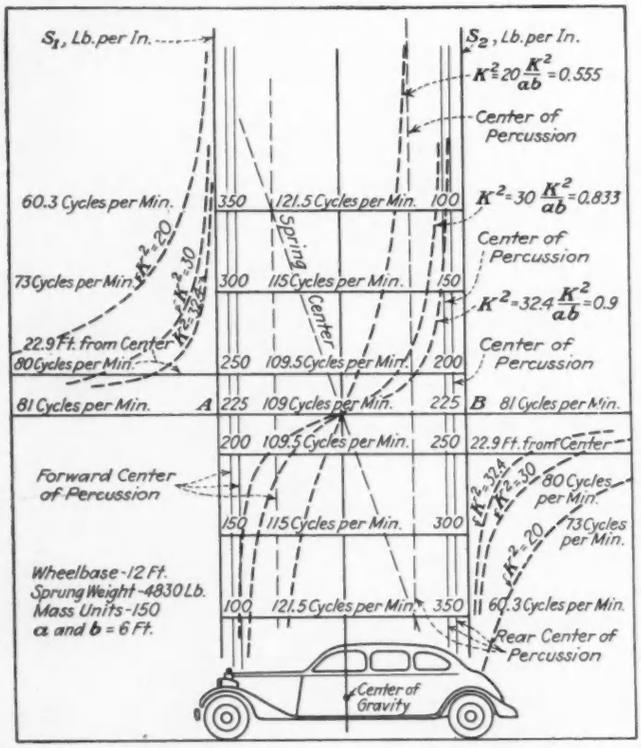


Fig. 5—Shift of Oscillation Centers with Changes in Spring Rates

springs are softened the pitch slows down rapidly, the bounce more slowly, but since both centers are moving forward the rear passengers are more "out on a limb" from both of them.

The "boulevard" ride improves as the springs are softened (provided that damping and frictional effects in the front suspension are reduced at the same time) but the action of the car at speed, does not immediately improve. The car tends to throw more at the rear at high speeds. However, if we persist in softening the front springs the ride eventually starts to level out. This occurs as the front deflection approaches equality with the rear deflection.

Equal Deflections. Parallel Springing

By the time we have made front deflection equal to rear, the pitch center has crawled forward to the center of gravity, while the bounce center has moved off to infinity, i. e., the bounce has become parallel.

With one of the two frequencies parallel the only horizontal fore-and-aft motion of high points in the car, such as the passengers' heads, must be pure, simple, harmonic motion, free from any of the interference kicks or "heterodyning" shown in Fig. 6. At the higher speeds the tendency to

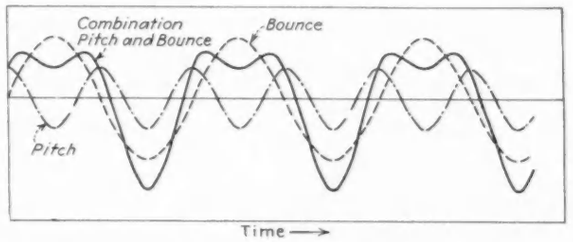


Fig. 6—Interference Kicks or "Heterodyning" of Ride in Car with Conventional Front Axle

absolutely flat parallel motion is most marked. Besides this improvement in action, the frequencies of the car are in their best relation, the ratio of pitch to bounce frequency being at its lowest possible value. From some viewpoints therefore this would seem to be the ideal ride. Actually it does not appear to be quite the best condition. Blows at either end of the car now fail to excite anything like a "pure" motion either of pitch or bounce. A mixture of almost equal proportions of pitch and bounce is caused, whichever end it struck, the two ends having similar effects. A road wave which will react forcibly on the front end will react in the same way on the rear. So that under certain road conditions at low road speeds the car will pick up a slow pitching action which, however, can be held within reasonable limits by quite a soft shock-absorber setting.

Now suppose the front springs are softened still further. Both centers still move forward, but the bounce center is already at infinity, so it has to reappear *behind* the car and comes rapidly closer as the pitch center creeps forward ahead of the center of gravity. The front end becomes the end which chiefly excites bounce while the rear end is "the boss" for pitching. The bounce frequency is now the one which slows down more rapidly, the pitch frequency decreasing more and more slowly as it approaches the basic frequency of the rear end. The difference between the frequencies becomes greater and their ratio starts increasing again.

Fig. 7 shows these changes in frequency through the range of practical values.

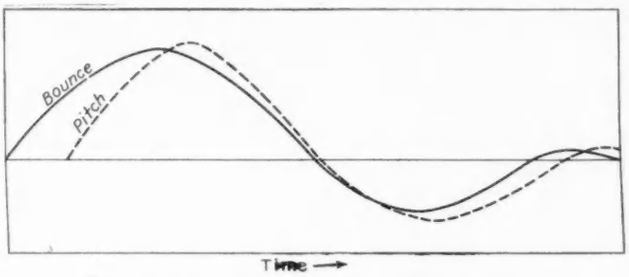


Fig. 8—Pitch and Bounce Sequence on Car with Reversed Springing

Some factors of ride are getting worse, but one important factor is getting better. The motion stirred up by the front end is becoming definitely slower than that started at the rear. (See Fig. 8)

On passing over an obstacle the car will still tend to swing to a level position and will hold it until the oscillation has nearly died out. Bear in mind that even at the lightest shock settings a very high degree of damping is always obtained in a motor car as compared with many of the other oscillating systems which haunt the lives of engineers, so that any motion of the car started by a single wave is soon damped out.

From what has been said it is obvious that there must be a range in the ratio of rear to front-spring deflection *below* the condition of equal deflection during which the action of the car will appear very nearly level. But if we go on softening the front springs too far relative to the rear we shall finally reach the stage where the front end appears to move over the road without vertical movement while the rear end will kick up all by itself; where, therefore, the impression of parallel ride will begin to disappear.

Now, due to varying passenger and fuel loads, if the average passenger car is sprung so that under absolute maximum loads the rear deflection is equal to front deflection, at the lightest load under which it can be driven the rear deflection will be equal to about 70 per cent of the front. A car sprung in this way comes the nearest to giving an impression of level riding under all conditions of load, road and speed.

There are other angles to this reasoning but they indicate that the front spring deflection should be equal to or somewhat greater than the rear and that on cars in which passenger load causes considerable variation in the rear deflection the springs should be so chosen that the front and rear deflections are equal only at absolute maximum load.

At normal traveling loads the front end will then be sprung from 10 to 20 per cent softer than the rear.

K^2/ab Ratio

Apparently no discussion of ride can be considered complete without some reference to the K^2/ab ratio. This is the ratio between the radius of gyration squared and the product of the two parts of the wheelbase either side of the center of gravity. It is a measure, divorced from scale effects, of the effectiveness of the car as a flywheel. It has generally been considered that if K^2/ab is equal to 1, with any spring-system, the best possible ride with that spring-system is obtained. On the conventionally sprung car this is certainly so, since both frequencies front and rear become "basic," i. e., 10 to 20 per cent lower than present values; their ratio becomes the lowest for that particular system of springing, the oscillation centers are actually at the axles, so that a front blow does not precompress the rear springs, etc.

We all know the improvement in ride in the conventional car when a heavy trunk is fitted at the rear, or when both spare wheels are taken off the side and overhung at the rear, and that this improvement exists even when rear springs are stiffened up to compensate for the increased load. But simply making $K^2/ab = 1$ without changing the springing does not get rid of vertical interference kicks of which the driver who sits near the middle of the car gets the most benefit, nor of horizontal interference kicks at the passenger's heads.

The ratio of $\frac{\text{Pitch frequency}}{\text{Bounce frequency}}$ under these conditions is equal to the square root of $\frac{\text{Rear deflection}}{\text{Front deflection}}$ so that in the case used before where we had 4 in. front deflection and 9 in. rear, the pitch would still be 50 per cent faster than bounce, which will give quite severe interference kicks. The question naturally arises "Why not do both? Why not make the spring deflections equal front and rear, and make $K^2 = ab$?"

This also we have tried. This is the condition where the car has no fixed oscillation centers and, therefore, no definite typical action. On the road this type of ride appeared to us pleasant but relatively difficult to control. It was not enough different from the results obtained with a K^2/ab ratio of 0.75 to justify the extremely peculiar appearance which a large car must have to obtain these conditions.

When the springing is parallel the chief importance of K^2/ab is that $\frac{\text{Pitch frequency}}{\text{Bounce frequency}} = \sqrt{\frac{ab}{K^2}}$ so that a K^2/ab ratio of 0.75 will give pitch only 15 per cent faster than bounce.

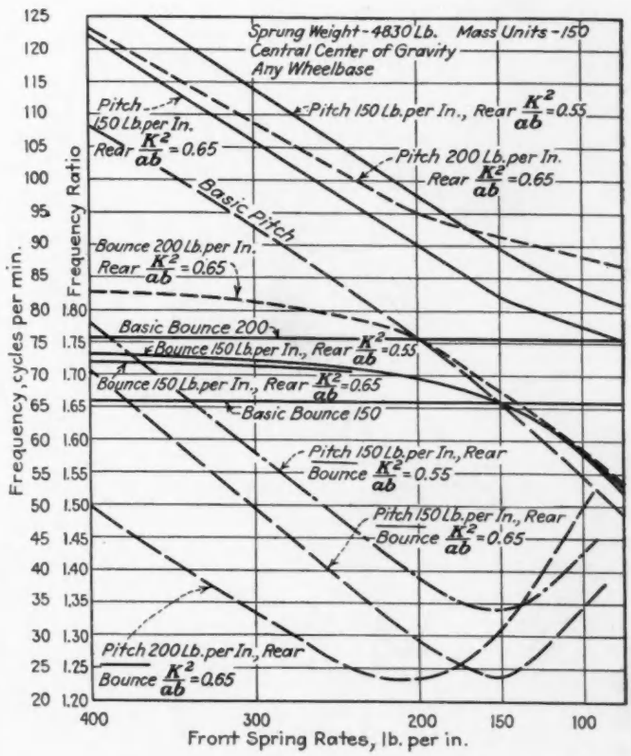

Sprung Weight - 4830 Lb. Mass Units - 150
Central Center of Gravity
Any Wheelbase
Pitch 150 Lb. per In., Rear K²/ab = 0.55
Pitch 150 Lb. per In. Rear K²/ab = 0.65
Pitch 200 Lb. per In. Rear K²/ab = 0.65
Basic Pitch
Bounce 200 Lb. per In. Rear K²/ab = 0.65
Basic Bounce 200
Bounce 150 Lb. per In., Rear K²/ab = 0.55
Bounce 150 Lb. per In., Rear K²/ab = 0.65
Basic Bounce 150
Pitch 150 Lb. per In., Rear Bounce K²/ab = 0.55
Pitch 150 Lb. per In., Rear Bounce K²/ab = 0.65
Pitch 200 Lb. per In., Rear Bounce K²/ab = 0.65
Frequency, cycles per min.
Frequency Ratio
Front Spring Rates, lb. per in.
125 120 115 110 105 100 95 90 85 80 75 70 65 60 55 50 45 40 35 30 25
1.80 1.75 1.70 1.65 1.60 1.55 1.50 1.45 1.40 1.35 1.30 1.25 1.20
400 300 200 150 100

Fig. 7—Frequency in Pitch and Bounce with Various Front Springs

The difference in actual road test between this and absolutely equal frequencies is so small that with one's eyes closed it is not easy to detect which condition is being used. This will be evident when one considers the high damping factor which always must exist in motor-car suspensions.

The car on which the first ride tests were made is shown in Fig. 9. This was a standard 7-passenger sedan deliberately chosen because of its considerable rear overhang. Outriggers were fitted front and rear and twelve 60-lb cast-iron weights were housed in a box in the rear compartment or clamped to the outriggers as desired. The outriggers were equi-distant from the box so that if the weights were taken out in pairs and clamped one to each outrigger, the moment of inertia was increased without changing the weight distribution. Each outrigger could hold all twelve weights if desired. So, the whole range of possibilities in weight distribution, moment of inertia and spring deflections front and rear could be explored very rapidly and any likely combinations could be checked over and over again.

Frequency Level

Before leaving ride we have to decide *how soft* to spring the car. As has been stated, similar "action" can be produced at any frequency level, but the ride becomes softer and more agreeable and the excitation from the road becomes less as the springing is softened. Finally the thing which decides this more than anything else is the proximity of the rear passengers to the rear springs.

In large cars, the 7-passenger body is still popular and will probably remain so even with increasing width of seats. A middle passenger in a wide seat lacks both vision of the road and any sense of privacy. The popularity of corner seats in European railroad travel is a case in point. In conventional cars, the rear passenger, even in spite of considerable overhang on some cars, has suffered discomfort not from the rear, but from the front springs. However, with the front springs softened enough to remove this trouble entirely, it was found that the stiffness of the rear springs began to take on new importance. This has been explained above since the rear springs begin to control not bounce but pitch. Therefore it was found that the rear springs could not be stiffened. So immediately we were faced with the need of softening the front suspension not 30 or 40 per cent but between 2½ and 3 times.

This, therefore, was our introduction to independent suspension.

Independent Suspension

We consider that the best way of getting this soft front suspension and at the same time retaining stability and improving precision of handling, is to adopt one of the types of independent suspension. Before going any further we ought to attempt a definition. As I see it the general definition is:

A suspension in which wheel movement relative to the car is the same in roll as it is in ride.

But to define what we are thinking of as an independent *front* suspension I would modify this as follows: *A suspension in which the wheels move up and down independently of each other, in paths which are invariable, and which maintain the wheel planes closely parallel to the mid-plane of the vehicle.*

Independent suspension can be discussed best by describing its six outstanding features:

(1) It permits very large spring deflections with the use of frictionless springs. As deflection increases damping and friction must decrease, especially friction. Otherwise the ride isn't soft, it is merely bumpy.

(2) "Knee action." Either wheel can pass over an obstacle without disturbing the path of the other wheel or throwing the car sideways.

(3) It permits the designing of an accurate steering mechanism. Obviously until the paths of the wheels relative to the car are invariable a really precise steering layout is impossible.

From what has been said about ride it is obvious that if the car is to ride level, the up-and-down movement of the wheels relative to the car on wavy roads will greatly increase. Up till now we have restrained this relative motion with stiff springs and heavy shock-absorber settings. If the front suspension is to be allowed to "breathe" a new start has to be made on steering accuracy. On a 12-ft. wheelbase car at 100 m.p.h., 1 deg. of error in the path of the front wheels relative to the rear, represents a lateral acceleration equal to gravity and will turn the car over.

(4) Even with extremely soft and practically frictionless springing we can get absolute freedom from shimmy. There is one important condition, however—the frame and body assembly must be stiff enough. The lateral rolling action of the wheels, on the type of independent suspension we are considering, is linked rigidly to the frame, so that the rolling frequency of the car is between 50 and 70 cycles per min. This is also the roll frequency of the wheels.

The lateral rolling of the wheel planes, produced by gyroscopic reaction from the outward swerve of the tires on the road, is an essential link in building up the work-cycle of shimmy. But in order that the cycle may build up, the frequency of the lateral roll of the wheels must be of the same order as the hop-frequency of the wheels (400 to 600 cycles per min.). In a front axle the frequencies are the same since the wheels are connected by a rigid member. In an independent the connection is broken by making the roll frequency about one-tenth of the hop-frequency. But, obviously, if the torsional frequency of the frame and body assembly is of the order of 600 cycles per min., a form of shimmy is still possible. So that for a successful independent suspension the assembly of the sprung masses must be "tuned up" to a fairly high frequency. Hence the absolute need of a stiff frame.

It may be objected that the frame might also be "tuned down" to a lower frequency than the 600 per min. of initial hopping and still prevent shimmy. In other words the frame might be softened. This may be so, but I am afraid if this is attempted the experience will be that although everything may go well on good and fair roads, a series of potholes or very rough roads which throw the wheels off the ground and so slow the hop-frequency to, say 400 per min., will produce shimmy in a very violent and unexpected manner. In practice it appears that the frequency of the frame "nose" ahead of the dash should be held to about 800 cycles per min. if shimmy is to be completely avoided.

Notice also that, whereas shimmy on an axle car is more likely to occur if the tires are soft, it will only tend to occur in an independent suspension if the tires are much too hard. This is obvious since harder tires increase the wheel hop-frequency.

(5) For a given spring deflection independent front suspension increases the roll-stability on corners. Some investigators have claimed an advantage of 2 to 1 for the indepen-

dent over the front axle, reasoning that the lever arm on the independent is twice as long but the spring base is twice as wide, and the stability goes up with the square of the spring base and down directly with the lever arm. But we find this an unsatisfactory statement because tests on complete cars show that conventional leaf-springs are about 50 per cent stiffer in roll than in straight up-and-down action. Also the axle itself rolls considerably on its tires, especially with low-pressure tires. Finally, the shift of the center of gravity as the car rolls introduces a cumulative effect into rolling which the above simple reasoning does not allow for.

The three lines in Fig. 10, represent a comparison between a front axle in which the spring is mounted on spherical shackles or any similar torsionally-free shackle, a front axle with the spring held in stiff shackles, and a parallel action independent with no external device or modification of action to reduce rolling.

With stiff springing, the roll angle is in the proportion 2.0, 1.33, 1.0 respectively. But at 7 in. deflection, the proportion has changed to 3.1, 1.5, 1.0, due to increased shift of the center of gravity. The independent, therefore, has about two-thirds of the roll angle of a conventional front axle with rigid shackles when spring deflections are used of the order of 7 in. The advantage increases as the suspension is made softer.

It has been asked why torsional stabilizers are being used on some of the new cars with independent front suspension. One answer to this is given later in connection with steering. But generally it should be pointed out that the overall softness of the suspension has been increased about 50 per cent, that is, the "deflection" measured at the center of gravity of the car has been increased this much. Reference to the chart shows that an increase of 50 per cent in spring deflection means an increase in roll angle of about 75 per cent. So the use of a torsion stabilizer is required not because of the use of independent suspension but because of the increased overall softness of the suspension.

(6) Inherent stability in handling. When an axle-car rolls in turning a corner the wheels remain more or less square with the road. When a parallel-action independent rolls on a corner the wheels roll with it. Here is one of the main problems, and (when it is properly controlled) one of the main advantages of independent front suspension.

First of all Fig. 11 is a general picture of a car making a turn at speed. The wheels steer the car around the corner like the rudder of a boat, developing the necessary centripetal force by virtue of the "slip angles" φ_1, and φ_2 between the wheel planes and their paths of travel. On a flat turn, therefore, at any speed above a walking pace and on a banked turn at any speed above the speed for which the curve is banked, the "Ackerman" layout which we work out in the drawing office no longer applies.

For example, on a car of 10-ft. wheelbase on a flat turn of 800 ft.-radius at 60 m.p.h., the slip angles φ_1 and φ_2 are about 4 deg., the center of the turn is not on the rear-axle center produced but is 46 ft. ahead of the front axle center, and the rear wheels will track 8 in. outside the front wheels. The first thing is to get to know something about the side thrusts produced by the slip angles. These thrusts are proportional to slip angle (up to about 5 deg.), independent of speed; independent of load on the wheel; increasing with tire pressure; increasing with tire size; and applied an inch or more *back* of the center of the tire contact patch. They represent, therefore, a self-straightening torque about the kingpin.

The torque about the kingpin has even more peculiar characteristics than the side thrust, chief of which is that though the side thrust is independent of wheel load the torque increases directly with wheel load.

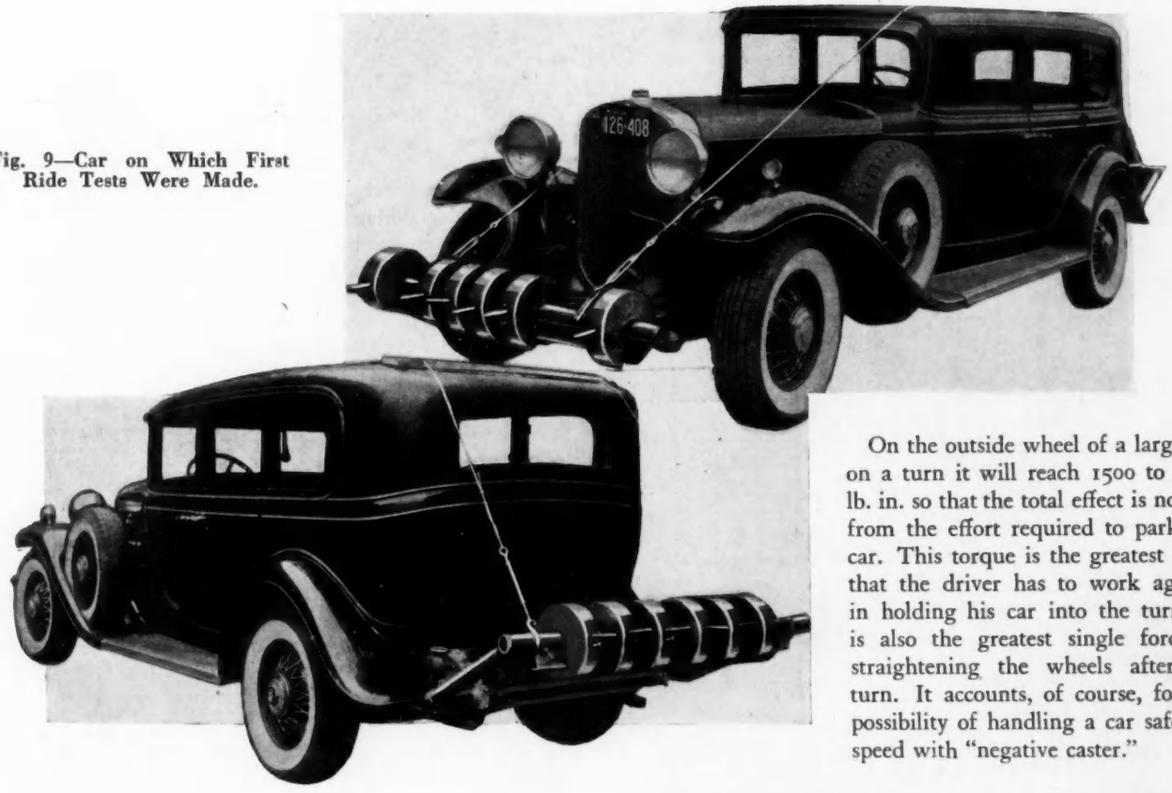


Fig. 9—Car on Which First Ride Tests Were Made.

On the outside wheel of a large car on a turn it will reach 1500 to 1600 lb. in. so that the total effect is not far from the effort required to park the car. This torque is the greatest force that the driver has to work against in holding his car into the turn. It is also the greatest single force in straightening the wheels after the turn. It accounts, of course, for the possibility of handling a car safely at speed with "negative caster."

On a parallel action independent, the wheels roll with the car. To put it a little more exactly, the wheel planes tend to take up a slightly acute angle with the plane of the road towards the outside of the turn. The effect of changing the camber of a rolling wheel is to produce a thrust in the direction of the acute angle. The center of application of this force is slightly *ahead* of the center of tire contact. So this camber force also tends to straighten the wheels. Since it acts in the opposite direction to the side thrust from slip angle, its effect is to slightly reduce the centripetal force acting on the car. (See Fig. 12.)

The camber-thrust per degree is about one-sixth of the thrust due to slip angle at normal load, but the camber-thrust is proportional to the load on the wheel and increases with a decrease in tire pressure, so the relation is not constant.

From what has been said it is clear that when a car with independent front wheels rolls slightly in making a turn, the front slip angle will tend to increase relative to the rear; in other words, the radius of the turn will increase, or the curve will "flatten." At the same time, as the car rolls, there will be a slight increase in the torque at the steering wheel tending to straighten up the wheels. Or suppose the car is going straight on a cambered road and rolls slightly in passing over the crest of the camber. The tendency of an independent front end is to make the car steer "in under" its roll, like a bicycle. We have called this "bicycle stability." It probably accounts for the incomparable handling of independents on wet or icy streets, frozen car-tracks, or in the event of tire bursts.

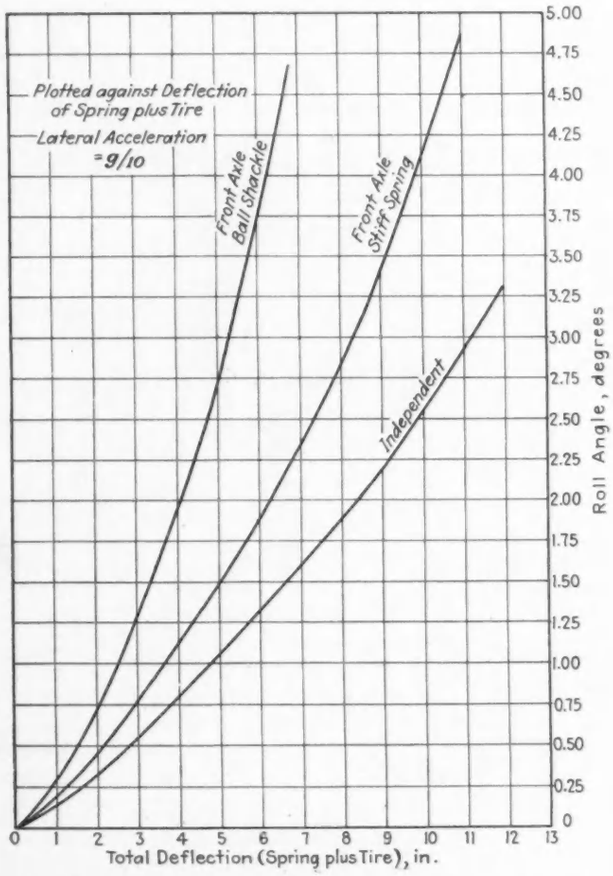


Fig. 10—Relative Instability on Corners of Various Types of Suspension

We have always tried to get a little of this effect in the steering geometry layouts of front axles, if only because the contrary effect, in which a right-hand roll produces a left-hand turn (and a left-hand roll a right-hand turn) is so highly dangerous.

Let us follow out this opposite effect, and suppose a car turning left to roll a little to the right. If the right-hand roll produces ever so slight an increase in the left-hand turn of the front wheels, this increases the sharpness of the curve followed by the car, which increases centrifugal force and right-hand roll, and a vicious circle of events is set up, which but for the vigilance of the driver will end up in a flat spin or an overturned car. I expect everybody has produced such steerings either accidentally or deliberately and knows how handling the car gives the impression of high-speed tightrope walking.

Producing stability in a front axle by "faking" the geometry in the stable direction had the disadvantage that this "faking" was apparent under conditions when we didn't want it; for example, when driving the car straight ahead on a road with transverse waves. This is not so with an independent, since the automatic stability on turns may be combined with a perfect steering geometry.

Scale Effects

The thrust due to camber angle and the increased torque about the kingpin produced by wheel camber appear to increase as the load on the wheel increases. Furthermore, we have found no way to avoid putting higher reduction ratios in the steering gear of large cars than in small cars, although for equal facility of handling it is obvious that the reduction ratio should vary inversely with the length of the wheelbase—if a 120-in. wheelbase has a steering ratio of 15 to 1, a 150-in. wheelbase should have a ratio of 12 to 1 to get around the same corner with the same angle of turn of the steering wheel. So we should expect these "stability effects," due to the parallel action of the front wheels, which are pure gain in small cars, to call for careful examination and perhaps to require modifying on the larger vehicles. This does not seem to be anything to worry about, since it is a general rule in engineering and in nature that to be successful a design has to be made "fussier" as the scale is increased. In this particular case the solutions are fairly obvious: (a) to control rolling, by torsional stabilizers or other means; (b) to lose wheel camber, that is, camber the wheels inward, as they rise, so that the outer, or more heavily loaded wheel on a corner shall remain more nearly square with the road; and (c) to lose caster as the wheels rise so as to prevent excessive increase in the self-centering effect of the wheels.

We believe all or any of these are useful devices to adopt for the improved handling of the heavier cars.

Tire Wear

Questions have been raised on the wear of tires with independent suspension. I think the reason is that there have been two methods of designing independents. One way has been to make a design that looks as though it ought to cause tire wear, find that it did, and then call on the tire makers for help. The other has been to work with the tire makers to find what a design ought to be to avoid tire wear.

From what has gone before it is obvious that if the car rolls on a corner and if the wheels roll as much as the car, the wheels must run at a greater slip angle to make a turn of a given radius. Under these conditions, therefore, we might expect more tire wear. But turning corners of 800-ft.

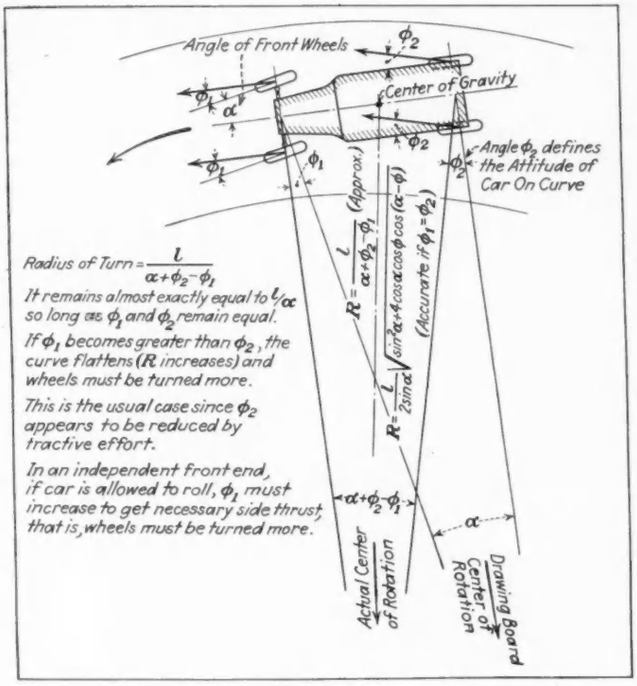

Angle of Front Wheels
Center of Gravity
Angle ϕ_2 defines the Attitude of Car On Curve
Radius of Turn = $\frac{l}{\alpha+\phi_2-\phi_1}$
It remains almost exactly equal to l/α so long as ϕ_1 and ϕ_2 remain equal.
If ϕ_1 becomes greater than ϕ_2, the curve flattens (R increases) and wheels must be turned more.
This is the usual case since ϕ_2 appears to be reduced by tractive effort.
In an independent front end, if car is allowed to roll, ϕ_1 must increase to get necessary side thrust, that is, wheels must be turned more.
$R = \frac{l}{\alpha+\phi_2-\phi_1}$ (Approx.)
$R = \frac{l}{2\sin\alpha}\sqrt{\sin^2\alpha+4\cos\alpha\cos\phi\cos(\alpha-\phi)}$ (Accurate if $\phi_1=\phi_2$)
Actual Center of Rotation
Drawing Board Center of Rotation

Fig. 11—High-Speed Turns

radius at 60 m.p.h. does not occupy a great percentage of the running life of the average passenger car, and under other conditions—for example, when the wheels are surmounting obstacles alternately—the wear of tires on an independent is less, since there is little or no scrubbing of the tire.

As far as we know, no modification of tire design to meet independent suspensions has had to be made except in cases where the manufacturer was changing to a lower pressure tire equipment.

In parallel road tests with identical tire equipment on independent and axle cars no difference in wear has been detected.

Coil Springs

Last fall, from steel makers who had not supplied us with spring steel, we first heard of spring breakage. As far as we can find out, the breakage is almost purely imaginary. We at Cadillac having failed to produce spring breakage on the road, finally built an oscillating testing machine consisting of two of our standard lower "wishbones" alternately compressing a pair of springs which rest on a stationary block between them. This tests the springs through full working range under exactly the conditions of use, which is more than one can say of the usual leaf-spring test. The life on this test is twice the life of leaf springs.

On the road, all we can say is that it will be proportionately very much more, partly because of the lower average working stress range on such soft springs, and partly because the spring has no "offside" duties to perform, such as resisting brake torque, resisting roll by torsion, etc. If the coil spring should break, it will have no effect in the control of the wheel, except that the car will sit down on the large rubber bumper provided. Springs are silico-manganese No. 9260, which has so far proved the best for the purpose.

Change of camber of the wheels has sometimes been mentioned against the "wishbone" type of suspension because of the supposed effect in setting up gyroscopic forces. It should be obvious, however that, though this design cambers the wheels inward as they rise and fall, there is no *rate of change of camber* at the normal load position when the arms are horizontal. Therefore, there is no gyroscopic reaction at this position, and no comparison with a front axle under similar conditions of movement.

When testing these cars on large drums fitted with cams simulating a washboard road, and with the shock-absorber effect reduced to a minimum, we have made the wheels jump 3 in. into the air without a sign of wheel wabble. No such results can ever be obtained in our experience with a conventional front axle.

Conclusion

In discussing the reasons for independent suspension I have tried to show how these designs came about as an essential step for improving the riding and handling of cars. Without these improvements we consider that independent suspension would have no meaning. "Knee action," which for its full realization depends on soft suspension, we regard as just one phase of the improvement in ride.

The next question is "where do we go from here," and I imagine the answer to that is, if we live long enough we shall find out. Apologies may be due for throwing out a number of statements in the course of this paper without full explanation. The excuse for this is that the subject is wide, representing three years of pretty hard work, and in the time available it has only been possible to give a summary of the results obtained.

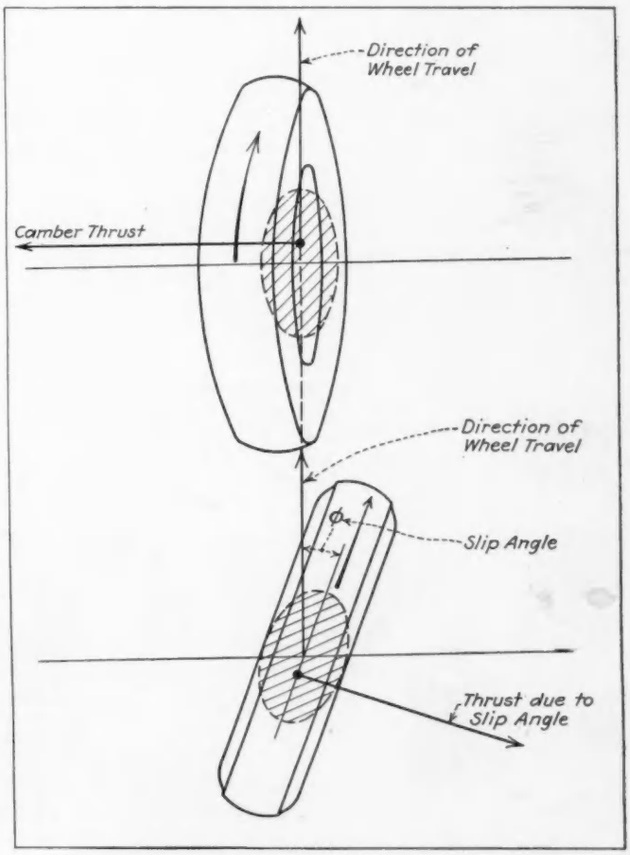


Fig. 12—Thrusts Due to Camber and Slip Angle

Aerodynamics of Wing Sections for Airplanes

By Eastman N. Jacobs

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National Advisory Committee for Aeronautics

IN this paper the aerodynamics of airfoil sections will be treated according to physical, rather than theoretical, considerations; that is, the actual air-flow, not the theoretical potential flow, about airfoil sections will be studied and also the resulting air forces acting on the sections. Information that may be obtained from such a study will be of direct value to engineers who are concerned with practical problems relating to the aerodynamics of airfoils.

The theoretical potential-flow calculations indicate that theoretical airfoil sections, even very thick ones, have no drag (profile drag) and no maximum lift in the sense that actual sections have a maximum lift as the result of burbling. The theory therefore gives little information on which to base the choice of airfoil sections except in regard to the pitching-moment characteristics, which are given approximately below the stall. The most important requirements of an airfoil section are that it develop a force perpendicular to its motion through the air (lift), little force in the direction of motion (drag), and provide adequate room within the section for an efficient structure. In these respects the theory gives nothing to choose between the sections; perfect results are obtained when any thickness is chosen and any shape, provided that the section has a rounded nose and a sharp tail. It is therefore apparent that the choice of suitable wing sections must depend largely on the character of the departure of the actual from the theoretical behavior.

The Theoretical Flow.—In the theoretical potential flow, the air forces are transmitted to the section entirely by air pressures acting on the surface. An airfoil section is so shaped that the air in flowing around it and leaving it at the sharp trailing edge creates a reduced pressure above and an increased pressure below the section. The low pressure over the upper surface may be thought of as existing in order to provide the pressure gradient supplying the centrifugal force required to cause the air to follow the curved paths over the section. The reduced pressure above the section and the increased pressure below give rise to the lift. The lift must act in the direction perpendicular to the motion since a force acting in any other direction would do work that cannot be accounted for in the flow.

[This paper was presented at the International Automotive Engineering Congress, Chicago, Sept. 4, 1933.]

[1] The smoke tunnel is usually operated at speeds between 3 and 5 ft. per sec. The airfoil chord is 10 in., thus the Reynolds Number range is approximately from 16,000 to 26,000.

The Actual Flow.—The characteristics of the actual flow differ from those of the theoretical mainly in that forces are transmitted to the body not only by means of pressures acting on its surface but by shearing stresses resulting from the fluid viscosity. That the layer of air immediately adjacent to the surface does not slip over the surface is considered an experimentally established fact. From this fact it follows that, regardless of how low the viscosity, a layer of air must exist adjacent to the body in which the viscous forces are comparable with the pressure forces. The layer in which these viscous shearing stresses are of importance is referred to as the boundary layer, following Prandtl to whom we owe much of our understanding of the mechanism of air-flows. The tangential force transmitted to the body surface is usually referred to as the skin friction.

Although the viscosity of air is relatively low and the skin-friction forces as compared with the pressure forces are often relatively insignificant, they may, however, become the controlling factor and completely alter the configuration of the flow pattern and the resulting air forces. Furthermore, as previously pointed out, these forces must determine the differences between the various airfoil sections on which the choice of an airfoil section must be based; that is, the action of viscosity must determine the drag and maximum lift of the section.

The importance of the viscous forces in determining the character of the air-flow about an airfoil section may best be appreciated by referring to Fig. 1 which is a photograph of the air-flow about an airfoil section at a relatively small angle of attack made by means of the N.A.C.A. smoke tunnel[1]. It is obvious that the flow pattern is altogether different from that given by the theory, the theoretical pattern following the surface and closing in smoothly behind the trailing edge. The actual flow, however, is observed to separate from the upper surface well forward of the trailing edge, leaving a region filled with relatively dead air over the after portion of the upper surface. This phenomenon of the boundary layer leaving the body surface is known as separation. It results from the viscous shearing forces and largely controls the extent to which the actual flow departs from the theoretical. The phenomenon of separation will be considered in detail later, but first it is necessary to consider the viscous shearing forces that produce it.

THIS paper, which has been awarded the Wright Brothers Medal for 1933, considers first the air forces acting on airfoil sections, their origin, and the nature of the differences between the air forces acting on an ideal airfoil section and on an actual one. These differences, which are due primarily to the action of viscosity, are considered in detail because they distinguish the desirable from the undesirable sections.

The effects of viscosity are discussed in relation to the air-flow about the section as affected by a variation of the dynamic scale or Reynolds Number of the flow. Separation of the flow from the airfoil surface, which produces the most marked deviations from the ideal flow, is discussed in detail, considering the effects of varying the Reynolds Number and the initial turbulence of the air.

Finally, the paper considers the variation of the aerodynamic properties of sections with changes in the section shape. These results are given from an analysis of the data from tests of 78 related airfoils in the N.A.C.A. variable-density wind tunnel at one value of the Reynolds Number. They provide the information required to make a more rational choice than heretofore possible of the most suitable section for a given application.

The Viscous Forces.—From the definition of the coefficient of viscosity μ the viscous shearing force on a small area l^2 of a surface may be written

$$F_v = \mu \left(\frac{dV}{dy}\right)_o l^2$$

where V is the velocity along the surface, y the distance from the surface, and the subscript o denotes the gradient at the surface. In two similar flows denoted by the subscripts 1 and 2 the ratio of the viscous forces on corresponding areas must therefore be

$$\frac{F_{v_1}}{F_{v_2}} = \frac{\mu_1 \frac{V_1}{l_1} l_1^2}{\mu_2 \frac{V_2}{l_2} l_2^2} = \frac{\mu_1 V_1 l_1}{\mu_2 V_2 l_2}$$

On the other hand, inertia forces in two similar flows must be in the ratio

$$\frac{F_1}{F_2} = \frac{\rho_1 V_1^2 l_1^2}{\rho_2 V_2^2 l_2^2}$$

where ρ is the density of the assumedly incompressible fluid. If the flows are to be similar the ratio of inertia to viscous forces must be the same in both

$$\frac{F_1}{F_{v_1}} = \frac{F_2}{F_{v_2}} = \frac{\rho_1 V_1^2 l_1^2}{\mu_1 V_1 l_1} = \frac{\rho_2 V_2^2 l_2^2}{\mu_2 V_2 l_2}$$

or

$$\frac{\rho_1 V_1 l_1}{\mu_1} = \frac{\rho_2 V_2 l_2}{\mu_2}$$

If l is taken as any characteristic length and the other terms are in any consistent system of units, the ratio $\frac{\rho V l}{\mu}$ is non-dimensional and is known as the Reynolds Number. The above equation is equivalent to stating that if the flows are to be similar the values of the Reynolds Number must be equal. Furthermore, the Reynolds Number may be considered as a measure of the ratio of the inertia or pressure forces to the viscous forces in a flow; with increasing values, the viscous forces become relatively smaller.

Separation.—In the initial stages of the flow about an airfoil section the phenomenon of separation is not observed. That is, as the air is accelerated from rest past the airfoil, or vice versa, the flow at first follows the airfoil surface as does the theoretical flow. Particles of air on approaching the upper airfoil surface are accelerated under the action of the pressure gradient acting toward the low-pressure region on the upper surface of the airfoil, a particle acquiring just sufficient kinetic energy in so doing to overcome the adverse gradients that act on it from this point aft. The air near the surface, however, loses some of its energy as the result of the viscous skin-friction drag and, lacking sufficient kinetic energy to overcome the adverse pressure gradients, is brought to rest before reaching the trailing edge of the airfoil. Such air tends to accumulate along the after part of the upper surface and to be returned toward the low-pressure point by the pressure gradient acting in that direction. Thus eventually a mass of dead air accumulates and is forced forward, by the action of pressure gradients, under the overrunning boundary layer producing finally the stable flow pattern with separation as shown in Fig. 1.

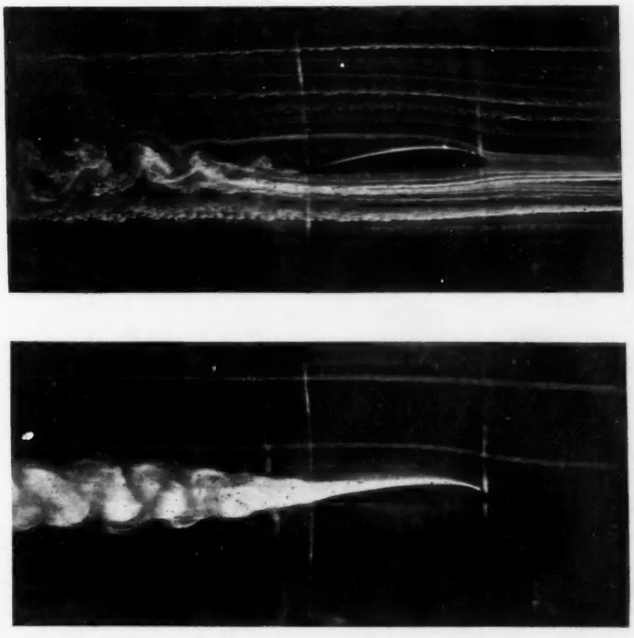


Fig. 1—Air-flow about an airfoil section at a small angle of attack

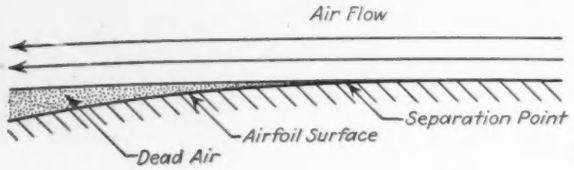


Fig. 2—Air-flow at the separation point

Consider now the equilibrium at the point of separation. The forward portion of the wedge of dead air, as shown in Fig. 2, is acted on by a pressure force tending to carry it forward under the overrunning boundary layer and a viscous shearing force tending to force it backward. These forces are in equilibrium. Consider now how this equilibrium condition is affected by increasing the Reynolds Number as, for example, by increasing the velocity of the air-flow. The pressure gradient tends to increase as the square of the velocity, whereas the skin-friction intensity increases less rapidly. We may therefore expect the separation point to move forward as the Reynolds Number is increased and to move aft as the Reynolds Number is decreased, finally disappearing entirely at very low values of the Reynolds Number corresponding, for example, to the slow movement of a small airfoil through molasses. From the above analysis, the following changes may be predicted as the Reynolds Number of the flow over an airfoil section is increased from very low values. The direct viscous skin-friction drag coefficient may be expected to decrease, the indirect pressure-drag coefficient may be expected to increase, and the maximum lift coefficient may be expected to decrease. These predictions must be modified, however, at higher values of the Reynolds Number as the result of the appearance of another phenomenon heretofore not considered, that of turbulence.

Turbulence.—The British physicist, Osborne Reynolds, for whom the Reynolds Number was named, first showed that the flow in a tube becomes turbulent at a certain value of that number dependent upon the initial turbulence of the fluid on entering the tube. Likewise, turbulence begins in other flows when the Reynolds Number exceeds a certain value. The onset of turbulence may be seen in Fig. 1 along the boundary layer behind the separation point. Considering it from the physical standpoint, the turbulence begins with the formation of small waves in the boundary layer, as shown in Fig. 3. As indicated in the upper part of the figure, the pressures produced by the overrunning air flowing over the crests of the waves produce in turn movements in the dead air tending to increase the amplitude of the waves and finally, as indicated, to produce distinct eddies, or turbulence. As in Reynolds' original experiments, the onset of such turbulence is dependent primarily on the value of the Reynolds Number, occurring sooner at high values of the Reynolds Number and not at all at very low values. Furthermore, the onset of this type of eddying flow is influenced by initial turbulence or unsteadiness in the air approaching the body.

Referring again to Fig. 1, the effect of the turbulence is to limit the extent of the dead-air region behind the airfoil. Dead air is constantly being carried away by the mixing of dead and live air in the eddies that pass off downstream. Prandtl has treated the action of turbulence as equivalent to increased viscosity. Before the onset of turbulence, shearing stresses are transmitted across the boundary layer by virtue of the viscosity, that is, by the transfer of momentum by the molecular movement in the air; whereas after the onset of turbulence an apparent increased viscosity results from the greatly augmented transfer of momentum as the result of lateral movement and mixing of the air in the eddies.

Now consider the air-flow about an airfoil section, such as that in Fig. 1, as affected by turbulence. At lower values of the Reynolds Number the eddies form so far behind the airfoil, if they form at all, that they have little effect on the character of flow near the airfoil. As the Reynolds Number is increased, the type of flow observed in Fig. 1 is established for which the extent of the dead-air region

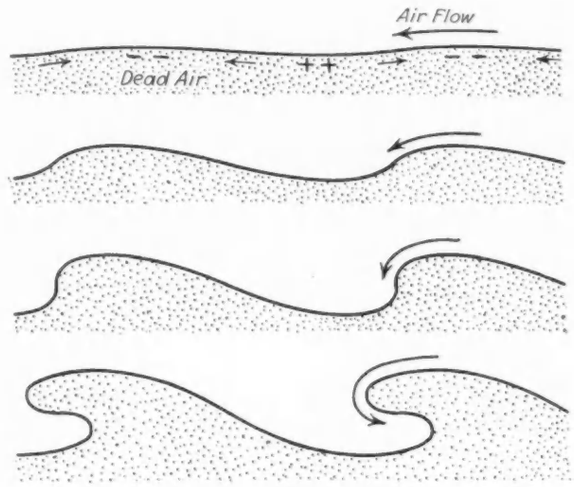


Fig. 3—Development of turbulence

is limited by the turbulence. As the Reynolds Number is further increased, the earlier onset of turbulence may be expected to reduce further the extent of the dead-air region, thus increasing the maximum lift coefficient and reducing the pressure drag coefficient. The skin-friction drag coefficient may be increased, however, as the result of the scouring action of the turbulent air in the eddies if they extend nearly to the surface of the airfoil. The action of turbulence in reducing the extent of the dead-air region behind an airfoil and consequently in increasing the maximum lift coefficient is illustrated in Fig. 4. Here the turbulence is produced by means of the small auxiliary airfoil ahead of the main one.

Now as the Reynolds Number is still further increased, turbulence may be expected to occur at or ahead of the separation point (see Fig. 2), upsetting entirely the equilibrium at this point between the pressure and the shearing forces. The scouring action of the turbulent air in the boundary layer may be expected to carry away the wedge of dead air[2] indicated in Fig. 2.

Initial Turbulence.—With increased turbulence or unsteadiness in the air approaching the airfoil, the boundary layer turbulence and the effects previously discussed may be expected to occur earlier; that is, at a lower value of the Reynolds Number. This conclusion is substantiated by earlier

[2] After the boundary layer at or just ahead of the incipient separation point has become turbulent, it is probable that further increases of the Reynolds Number produce relatively smaller shearing stresses and less resistance to flow breakdown, thus the adverse scale effect on the maximum lift of certain thick highly cambered airfoils for which the incipient separation point is well back from the leading edge corresponding to a relatively high local Reynolds Number.

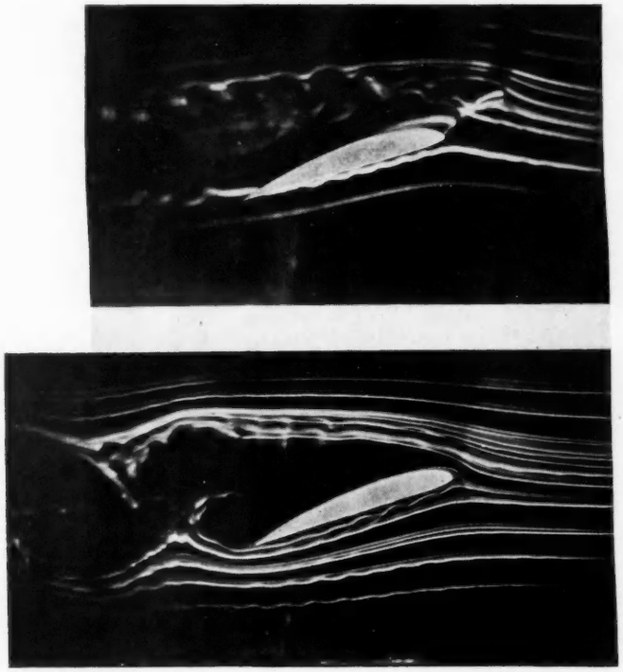


Fig. 4—Air-flow as affected by turbulence

experimental results[3] and by the results shown in Fig. 5. The curves represent the variation of the maximum lift coefficient of an airfoil with Reynolds Number as determined from tests in the N.A.C.A. full-scale tunnel and in the N.A.C.A. variable-density tunnel, where the high values of the Reynolds Number are obtained by increasing the pressure, or density, of the air in which the models are tested. The results from both tunnels, although they cannot be considered perfect on account of difficulties of measurement and influences produced by small variations of model form and surface conditions, indicate an increasing maximum lift coefficient with Reynolds Number within the full-scale range for airplane wing sections. The results from the variable-density tunnel, which has the larger initial air-stream turbulence, show the expected increase of maximum lift coefficient with Reynolds Number to occur at lower values of the Reynolds Number.

Airfoil Shape.—Nothing has been considered in regard to the effects of changing the shape of the airfoil section. In order to provide data on the variation of the aerodynamic characteristics with changes in the airfoil-section shape, the N.A.C.A. has measured the aerodynamic characteristics of a large number of related airfoils. For this investigation the variations of the characteristics with Reynolds Number and air-stream turbulence have been eliminated by making all the measurements under similar conditions; that is, those corresponding to a test in the N.A.C.A. variable-density tunnel of a standard size airfoil under normally large dynamic-scale test conditions. Although the airfoil characteristics may be expected to vary within the full-scale flight range of values of the Reynolds Number, the results of the investigation are considered typical within that range. The results have been published by the Committee[4]. It is proposed to present here a summary of the more important results.

Airfoil profiles may be considered as made up of certain profile thickness forms disposed about certain mean lines. The major shape variables then become two, the thickness form and the mean-line form. The thickness form is of particular importance from a structural standpoint. On the other hand, the form of the mean line determines almost independently some of the most important aerodynamic properties of the airfoil section; e. g., the angle of zero lift and the pitching-moment characteristics.

Description of Airfoils.—The related airfoil profiles for this investigation were derived by changing systematically these shape variables. The symmetrical profiles were defined in terms of a basic thickness variation, symmetrical airfoils of varying thickness being obtained by the application of factors to the basic ordinates. The cambered profiles were then developed by combining these thickness forms with various mean lines. The mean lines were obtained by varying the camber and by varying the shape of the mean line to alter the position of the maximum mean-line ordinate. *The maximum ordinate of the mean line is referred to as the camber of the airfoil and the position of the maximum ordinate of the mean line as the position of the camber. An airfoil, produced as described above, is designated by a number of four digits: the first indicates the camber in per cent of the chord; the second, the position of the camber in tenths of the chord from the leading edge; and the last two, the maximum thickness in per cent of the chord.* Thus the N.A.C.A. 2315 airfoil has a maximum camber of 2 per cent of the chord at a position 0.3 of the chord from the leading edge, and a maximum thickness of 15 per cent of the chord; the N.A.C.A. 0012 airfoil is a symmetrical airfoil having a maximum thickness of 12 per cent of the chord.

Well-known airfoils of a certain class including the Göttingen 398 and the Clark Y, which have proved to be efficient, are nearly alike when their camber is removed (mean line straightened) and they are reduced to the same maximum thickness. A thickness variation similar to that of these airfoils was therefore chosen for the development of the N.A.C.A. airfoils. An equation defining the shape was used as a method of producing fair profiles.

If the chord is taken along the x axis from 0 to 1, the ordinates y are given by an equation of the form

$$\pm y = a_0\sqrt{x} + a_1 x + a_2 x^2 + a_3 x^3 + a_4 x^4$$

The equation was adjusted to give the desired shape by

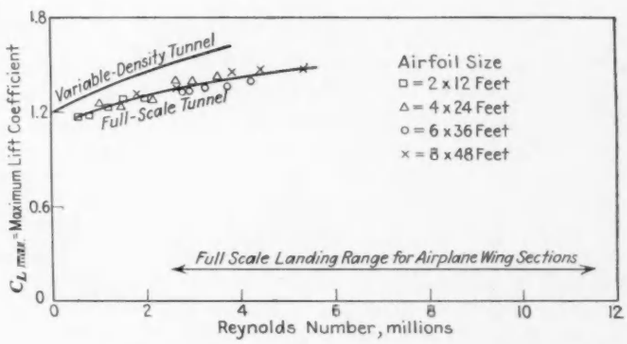
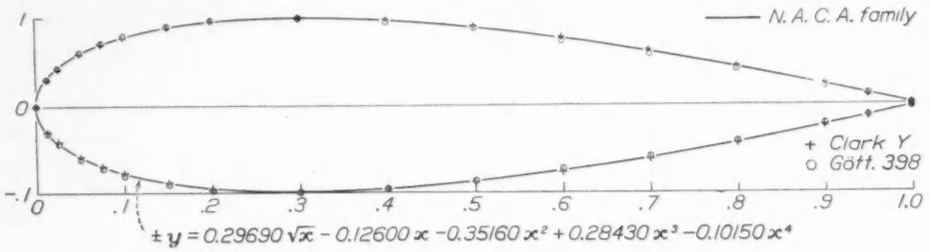


Fig. 5—Variation of $C_{L\ max.}$ with Reynolds Number

[3] Tests in the Variable Density Wind Tunnel to Investigate the Effects of Scale and Turbulence on Airfoil Characteristics, by John Stack; see N.A.C.A. 1931 Technical Note No. 364.

[4] The Characteristics of 78 Related Airfoil Sections from Tests in the Variable-Density Wind Tunnel, Eastman N. Jacobs, Kenneth E. Ward and Robert M. Pinkerton; see N.A.C.A. 1933 Technical Report No. 460.



Basic ordinates of N.A.C.A. family airfoils (per cent of chord)

Stations	0	1.25	2.5	5.0	7.5	10	15	20	25	30	40	50	60	70	80	90	95	100
Ordinates	0	3.157	4.358	5.925	7.000	7.805	8.909	9.563	9.902	10.003	9.672	8.823	7.606	6.107	4.372	2.413	1.344	0.210

L.E. radius, 4.40.

Fig. 6—Thickness variation

imposing the following conditions to determine the constants:

(1) Maximum ordinate 0.1 at 0.3 chord
$x=0.3 \quad y=0.1$
$dy/dx=0$

(2) Ordinate at trailing edge
$x=1 \quad y=0.002$

(3) Trailing-edge angle
$x=1 \quad dy/dx=-0.234$

(4) Nose shape
$x=0.1 \quad y=0.078$

The following equation satisfying approximately the above-mentioned conditions represents a profile having a thickness of approximately 20 per cent of the chord:

$$\pm y = 0.29690\sqrt{x} - 0.12600x - 0.35160x^2 + 0.28430x^3 - 0.10150x^4$$

This equation was taken to define the basic section. The basic profile and a table of ordinates are given in Fig. 6. Points obtained by removing the camber from the Göttingen 398 and the Clark Y sections, and applying a factor to the ordinates of the resulting thickness curves to bring them to the same maximum thickness, are plotted on the above figure for comparison. Sections having any desired maximum thickness were obtained by multiplying the basic ordinates by the proper factor; that is

$$\pm y_t = \frac{t}{0.20}(0.29690\sqrt{x} - 0\ 12600x - 0.35160x^2 + 0.28430x^3 - 0\ 10150x^4)$$

where t is the maximum thickness. The leading-edge radius is found to be

$$r_t = \frac{1}{2}\left(\frac{t}{0.20}a_e\right)^2 = 1.10t^2$$

When the mean lines of certain airfoils in common use were reduced to the same maximum ordinate and compared, it was found that their shapes were quite different. It was observed, however, that the range of shapes could be well covered by assuming some simple shape and varying the maximum ordinate and its position along the chord. The mean line was therefore arbitrarily defined by two parabolic equations of the form

$$y_c = b_0 + b_1x + b_2x^2$$

where the leading end of the mean line is at the origin and the trailing end is on the x axis at $x = 1$. The values of the constants for both equations were then expressed in terms of the above variables; namely,

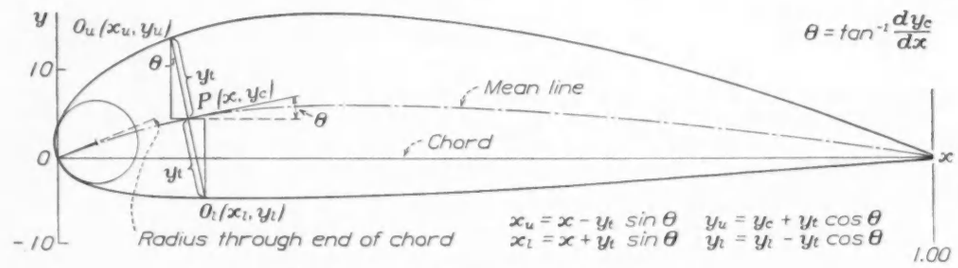
(1) Mean-line extremities
$x=o \quad y_c=0$
$x=1 \quad y_c=0$

(2) Maximum ordinate of mean line
$x=p$ (position of maximum ordinate)
$y_c=m$ (maximum ordinate)
$dy_c/dx=0$

The resulting equations defining the mean line then become

$$y_c = \frac{m}{p^2}[2px - x^2] \text{ (forward of maximum ordinate)}$$

and $$y_c = \frac{m}{(1-p)^2}[(1-2p) + 2px - x^2] \text{ (aft of maximum ordinate)}$$



Sample calculations for derivation of N.A.C.A. 6321

x	y_t	y_c	$\tan\theta$	$\sin\theta$	$\cos\theta$	$y_t\sin\theta$	$y_t\cos\theta$	x_u	y_u	x_l	y_l
0	0	0	[1] 0.40000	0.37140	0.92840	0	0	.....	.....	0	0
0.01250	0.03314	0.00489	0.38333	0.35793	0.93375	0.01186	0.03094	0.00064	0.03583	0.02436	—0.02605
0.30000	0.10503	0.06600	0	0	1	0	0.10503	0.30000	0.16503	0.30000	—0.04503
0.60000	0.07986	0.04898	—0 07347	—0.07327	0.99731	—0.00585	0.07965	0.60585	0.12863	0.59415	—0.03067
1	0.00221	0	—0.17143	—0.16897	0.98562	—0.00037	0.00218	1.00037	0.00218	0.99963	—0.00218

[1] Slope of radius through end of chord.

Fig. 7—Method of calculating ordinates of N.A.C.A. cambered airfoils

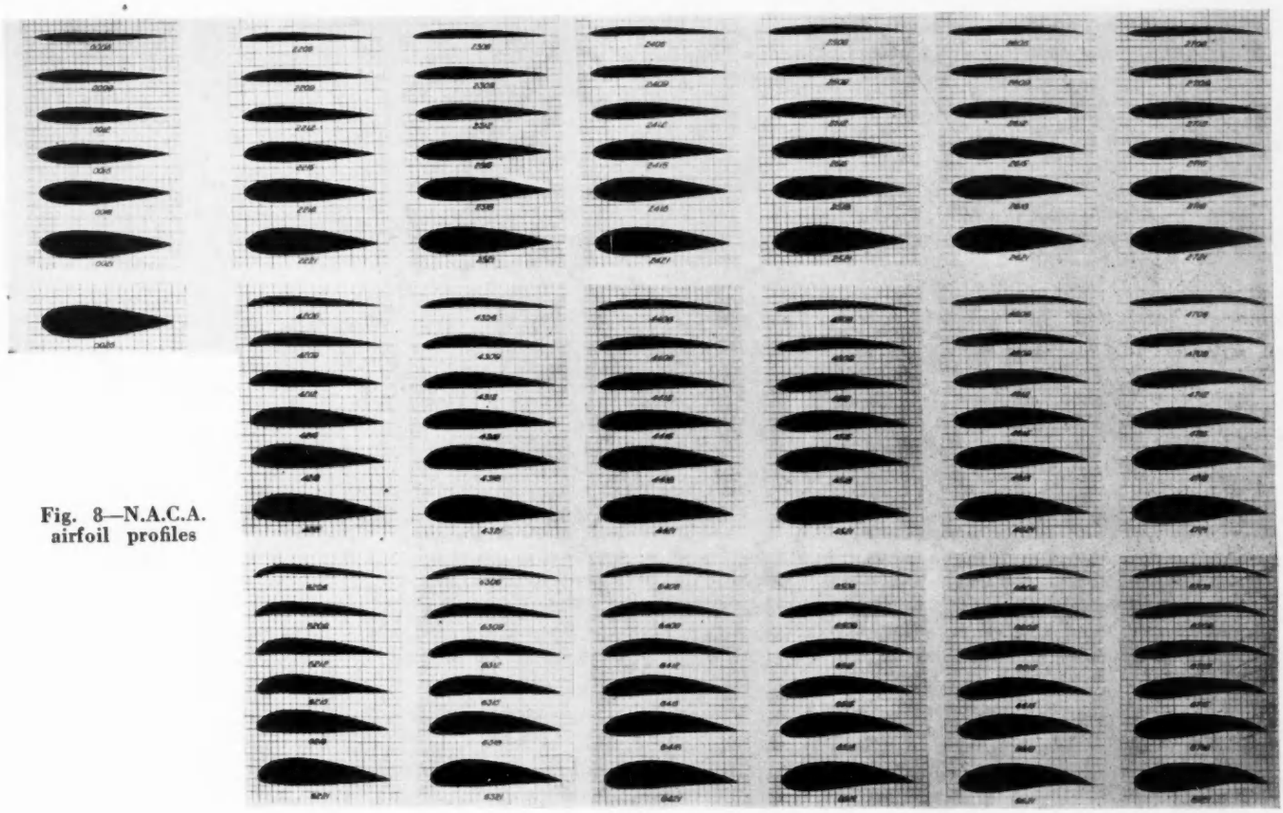


Fig. 8—N.A.C.A. airfoil profiles

The method of combining the thickness forms with the mean-line forms is best described by means of the diagram in Fig. 7. The line joining the extremities of the mean line is chosen as the chord. Referring to the diagram, the ordinate y_t of the thickness form is measured along the perpendicular to the mean line from a point on the mean line at the station along the chord corresponding to the value of x for which y_t was computed. The resulting upper and lower surface points are then designated:

Stations x_u and x_l

Ordinates y_u and y_l

where the subscripts u and l refer to upper and lower surfaces respectively. In addition to these symbols, the symbol θ is employed to designate the angle between the tangent to the mean line and the x axis. This angle is given by

$$\theta = \tan^{-1}\frac{dy_c}{dx}$$

The following formulas for calculating the ordinates may now be derived from the diagram:

$$x_u = x - y_t \sin\theta$$

$$y_u = y_c + y_t \cos\theta$$

$$x_l = x + y_t \sin\theta$$

$$y_l = y_c - y_t \cos\theta$$

Sample calculations are given in Fig. 7. The center for the leading-edge radius is placed on the tangent to the mean line at the leading edge.

A family of related airfoils was derived in the manner described. Seven values of the maximum thickness, 0.06, 0.09, 0.12, 0.15, 0.18, 0.21, and 0.25; four values of the camber, 0.00, 0.02, 0.04, and 0.06; and six values of the position of the camber, 0.2, 0.3, 0.4, 0.5, 0.6, and 0.7 were used to derive the related sections of this family. The profiles of the airfoils derived are shown collectively in Fig. 8. All the airfoils thus derived having a thickness of 0.12c were tested to investigate the effects of camber changes. Several groups having the same mean line were investigated first, however, to determine the effects of thickness changes. Thus, two sets of data which do not agree exactly are given for some of the sections of 0.12c thickness.

The results of a typical airfoil test are shown in Fig. 9. Such a figure contains all the data obtained from a standard airfoil test. The following discussion of the relationships mentioned earlier will be divided into two parts: the consideration, first, of the effects of thickness, and second, of the effects of camber.

Variation of Thickness.—In the usual working range of an airfoil section the lift coefficient may be expressed as a linear function of the angle of attack

$$C_L = a_o(\alpha_o - \alpha_{L_o})$$

where a_o is the slope of the lift curve for the wing of infinite aspect ratio and α_{L_o} is the angle of attack at zero lift.

The variation of the lift-curve slope with thickness is shown in Fig. 10. The points on the figure represent the deduced slopes as measured in the angular range of low profile drag. These results confirm previous results[5] in that they show the lift-curve slope to decrease with increasing thickness. It will be noted that all values of the slope lie below the approximate theoretical value for thin wings, 2π per radian; the measured values lie between 95 and 81 per cent, approximately, of the theoretical. Beyond this straight portion the slope falls off and finally a maximum value of the lift coefficient is reached. (See Fig. 9.) The value of the maximum lift coefficient is

[5] Large-Scale Aerodynamic Characteristics of Airfoils as Tested in the Variable-Density Wind Tunnel by Eastman N. Jacobs and Raymond F. Anderson; see N.A.C.A. 1930 Technical Report No. 352.

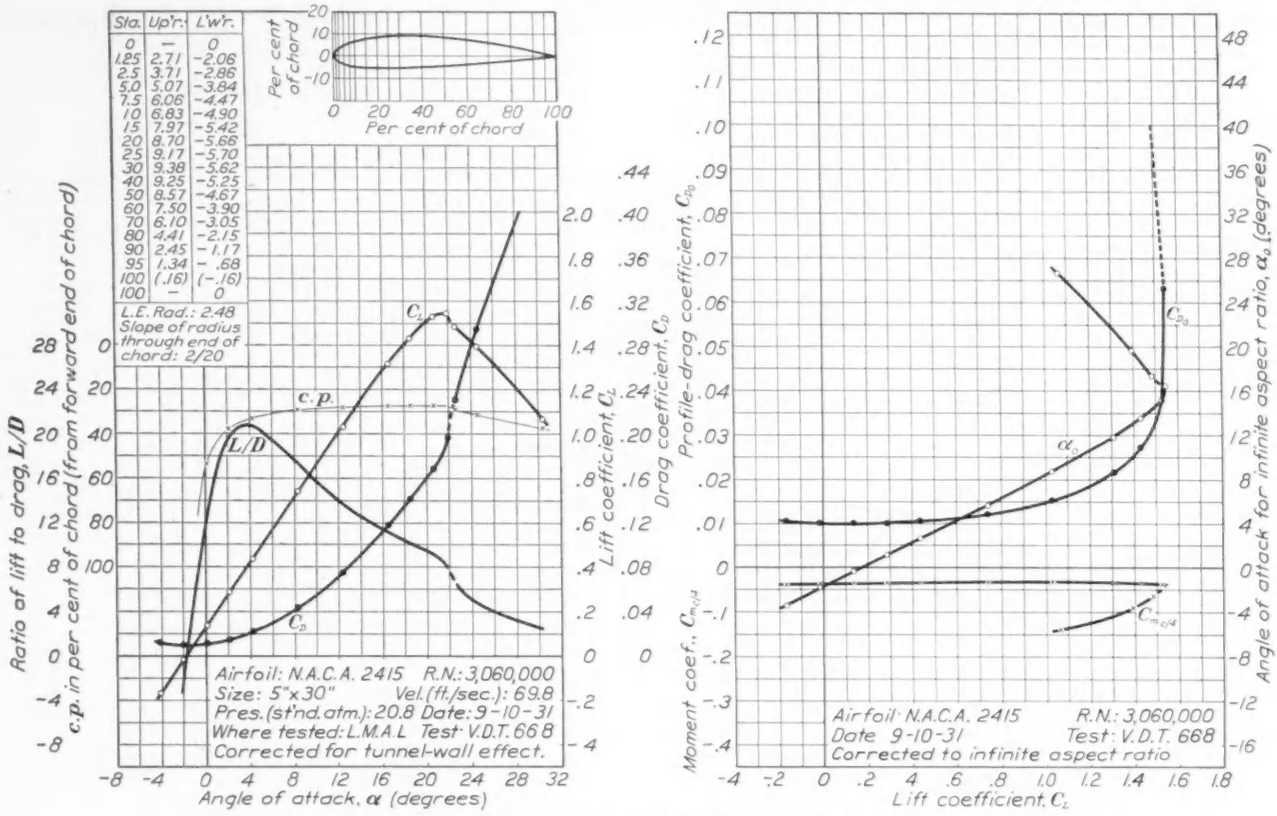

Sta. | Up'r. | L'w'r.
0 | — | 0
1.25 | 2.71 | -2.06
2.5 | 3.71 | -2.86
5.0 | 5.07 | -3.84
7.5 | 6.06 | -4.47
10 | 6.83 | -4.90
15 | 7.97 | -5.42
20 | 8.70 | -5.68
25 | 9.17 | -5.70
30 | 9.38 | -5.62
40 | 9.25 | -5.25
50 | 8.57 | -4.67
60 | 7.50 | -3.90
70 | 6.10 | -3.05
80 | 4.41 | -2.15
90 | 2.45 | -1.17
95 | 1.34 | - .68
100 | (.16) | (-.16)
100 | — | 0
L.E. Rad.: 2.48
Slope of radius through end of chord: 2/20
Per cent of chord
Per cent of chord
Ratio of lift to drag, L/D
c.p. in per cent of chord (from forward end of chord)
Lift coefficient, C_L
Drag coefficient, C_D
L/D
c.p.
C_L
C_D
Airfoil: N.A.C.A. 2415 R.N.: 3,060,000
Size: 5"x30" Vel.(ft./sec.): 69.8
Pres.(st'nd. atm.): 20.8 Date: 9-10-31
Where tested: L.M.A.L Test: V.D.T. 668
Corrected for tunnel-wall effect.
Angle of attack, α (degrees)
Profile-drag coefficient, C_{D_0}
Moment coef., $C_{m_{c/4}}$
Angle of attack for infinite aspect ratio, α_0 (degrees)
C_{D_0}
α_0
$C_{m_{c/4}}$
Airfoil: N.A.C.A. 2415 R.N.: 3,060,000
Date 9-10-31 Test: V.D.T. 668
Corrected to infinite aspect ratio
Lift coefficient, C_L

Fig. 9—Standard plot presenting results of airfoil tests in the variable-density tunnel

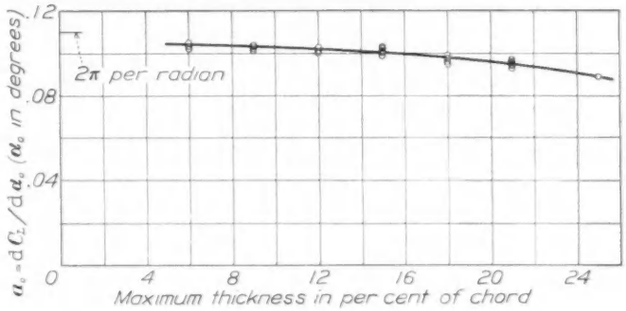


Fig. 10—*(at right)* Variation of lift-curve slope with thickness

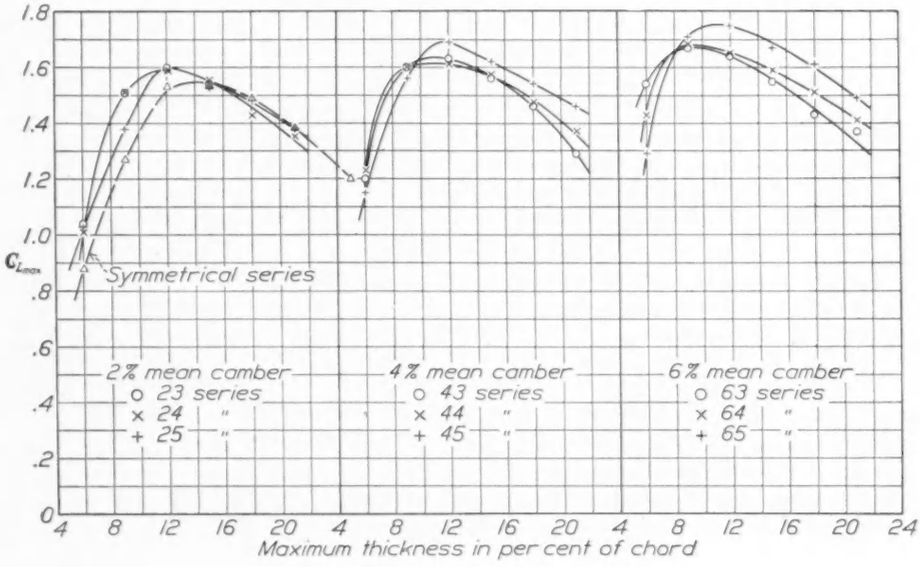


Fig. 11—Variation of maximum lift with thickness

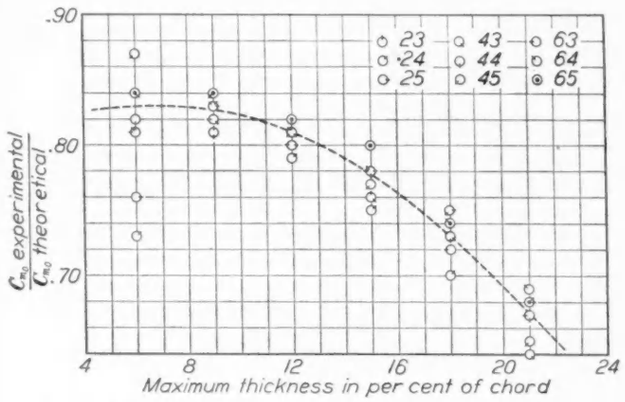


Fig. 12—Variation of moment at zero lift with thickness. Numbers refer to mean-camber designation

very greatly affected by changes in thickness as shown by Fig. 11. It will be noted that the highest values are obtained with moderately thick sections (9 to 12 per cent of the chord thick, except for the symmetrical sections for which the highest values are obtained with somewhat thicker sections).

Thin-airfoil theory (based on section mean line) separates the air forces acting on any airfoil into two parts: first, the forces that produce a couple but no lift (they are dependent only on the shape of the mean line); second, the forces that produce the lift only, the resultant of which acts at a fixed point. We then have in the working range an expression for the total moment taken about any point.

$$C_m = C_{m_o} + nC_L$$

[6] The N.A.C.A. Variable-Density Wind Tunnel, by Eastman N. Jacobs and Ira H. Abbott; see N.A.C.A. 1932 Technical Report No. 416.

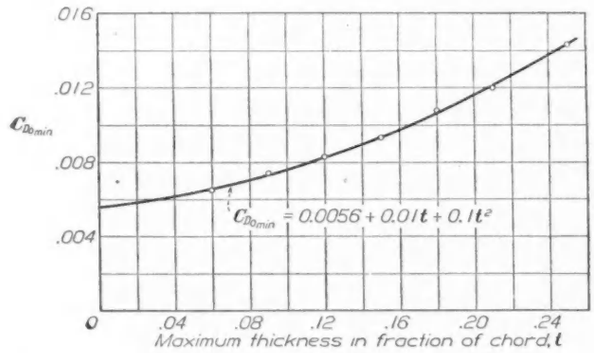


Fig. 14—Variation of minimum profile drag with thickness for the symmetrical airfoils

where C_{m_o} is the moment coefficient at zero lift and nC_L is the additional moment due to lift. Fig. 12 shows a decrease in the diving moment as the thickness increases. If the resultant of the lift forces acted exactly through the quarter-chord point as predicted by the theory of thin airfoils, there would be no additional moment due to the lift when the moments are taken about this point. The curves $C_{m_{c/4}}$ against C_L, however, show a slope in the working range which indicates that the axis of constant moment is displaced somewhat from the quarter-chord point. The factor n represents the amount of this displacement as obtained from the deduced slopes of the moment curves in the normal working range. Increasing the thickness moves the axis of constant moment forward, as shown by Fig. 13.

The total drag of an airfoil is considered as made up of the induced drag and the profile drag. Considering the profile

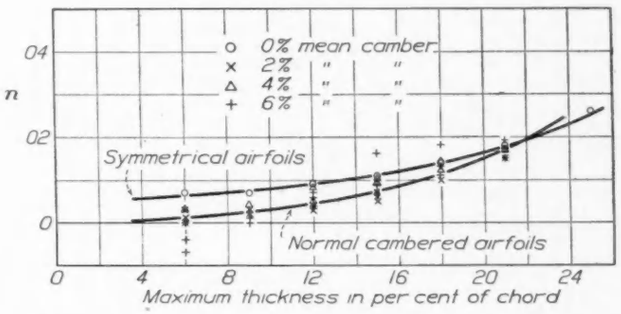


Fig. 13—Variation of position of constant moment with thickness. Values of n for equation

$$C_{m_{c/4}} = C_{m_o} + nC_L.$$

Results for airfoils having normal camber positions (0.3c to 0.5c).

drag as the minimum value plus an additional drag dependent upon the attitude of the airfoil, we have in coefficient form

$$C_D = C_{D_i} + (C_{D_{o\,min}} + \Delta C_{D_o})$$

The induced-drag coefficient C_{D_i}, which is computed by means of a previously published formula[6], is considered independent of the airfoil section. The variation of the profile-drag coefficient with the shape variables of the airfoil section is analyzed with respect to the variations of the two components of the profile drag.

The variation of the minimum profile-drag coefficient with thickness for the symmetrical sections is shown in Fig. 14. The cambered sections show the same general variation with thickness, but, to avoid confusion, the results are not plotted.

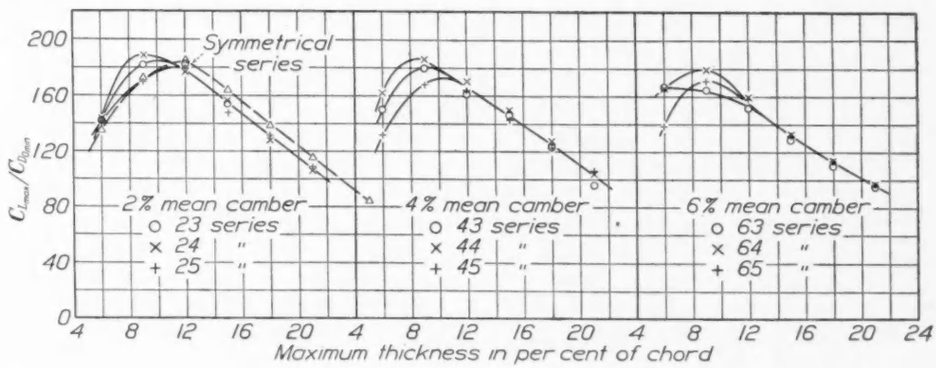


Fig. 15—Variation of $C_{L_{max}} / C_{D_{o\,min}}$ with thickness

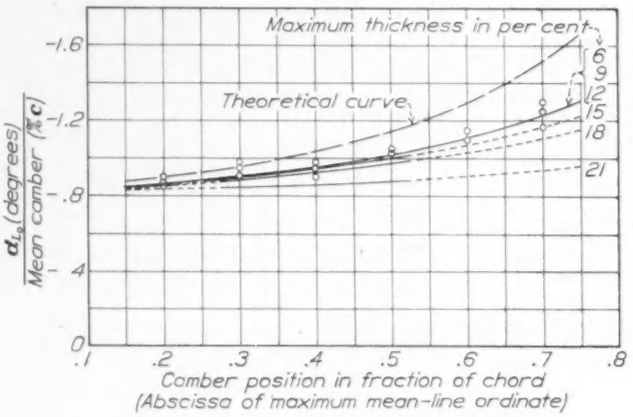


Fig. 16—Variation of angle of zero lift with camber. Points shown are for 12 per cent thick airfoils. Curves indicate general trends for the different thicknesses

The variation of the minimum profile-drag coefficient with the profile thickness may be expressed by the empirical relation

$$C_{D_{o\,min}} = k + 0.0056 + 0.01t + 0.1t^2$$

where t is the thickness ratio and k (which is approximately constant for sections having the same mean line) represents the increase in $C_{D_{o\,min}}$ above that computed for the symmetrical section of corresponding thickness.

The additional profile drag, which is dependent upon the attitude of the airfoil, has previously been expressed[7] as a function of the lift by the equation

$$\Delta C_{D_o} = C_{D_o} - C_{D_{o\,min}} = 0.0062\ (C_{L_{opt}})^2$$

where $C_{L_{opt}}$ may be called the optimum lift coefficient: that is, the lift coefficient corresponding to the minimum profile-drag coefficient. This equation holds reasonably well for the normally shaped airfoils at values of the lift coefficient below unity. It was found also that the value of the optimum lift decreased rapidly with increased thickness.

The general efficiency of an airfoil cannot be expressed by means of a single number. The ratio of the maximum lift to the minimum profile drag is, however, of some value as the measure of the efficiency of an airfoil section. The variation of this ratio with thickness is shown in Fig. 15. The curves of this figure indicate that the highest values of the ratio are given by the sections between 9 and 12 per cent of the chord thick.

Variation of Camber.—The effect of camber on the lift characteristics of an airfoil section is to make the angle of zero lift more negative and to increase the maximum lift coefficient. Thin-airfoil theory states that the angle of zero lift is proportional to the camber if the camber is varied as with these related airfoils by scaling the ordinates of a given mean line without altering the camber position. The theory also predicts an increased negative angle as the position of the camber moves back along the chord. The experimental values are compared with the theoretical values in Fig. 16.

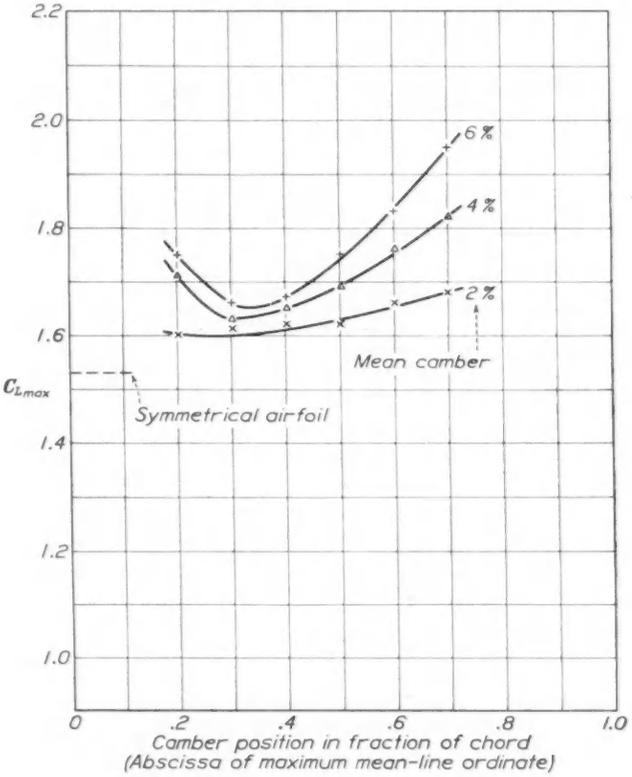


Fig. 17—Variation of maximum lift with camber. Results for 12 per cent thick airfoils

The variation of maximum lift coefficient with camber is shown in Fig. 17. The gain is small for the normal positions of the camber, but becomes larger as the camber moves either rearward or forward. It will be seen by reference to Fig. 11 that the camber becomes less effective as the thickness is increased.

The effect of camber on the drag is to increase the minimum profile-drag coefficient. The amount of this increase is given by k and the variation is shown in Fig. 18. As will be observed, the effect is not large except for highly cambered sections having the camber well back.

As with the angle of zero lift, the theory states that the moment at zero lift is proportional to the camber and predicts an increase in the magnitude of the moment as the camber

[7] Tests of N.A.C.A. Airfoils in the Variable-Density Wind Tunnel, Series 43 and 63, by Eastman N. Jacobs and Robert M. Pinkerton; see N.A.C.A. 1931 Technical Note No. 391.

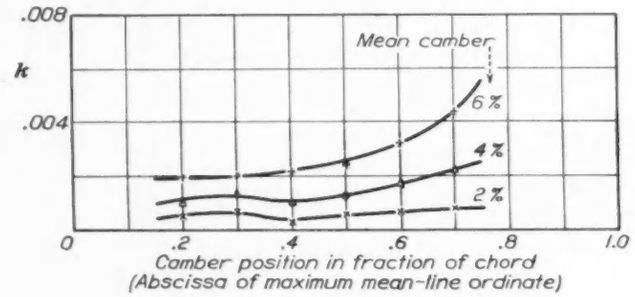


Fig. 18—Increase in minimum profile drag due to camber. Values of k for equation

$$C_{D_{o\,min}} = k + 0.0056 + 0.01t + 0.1t^2$$

k = increase in $C_{D_{o\,min}}$ due to camber

t = maximum thickness in fraction of chord. Results for 12 per cent thick airfoils.

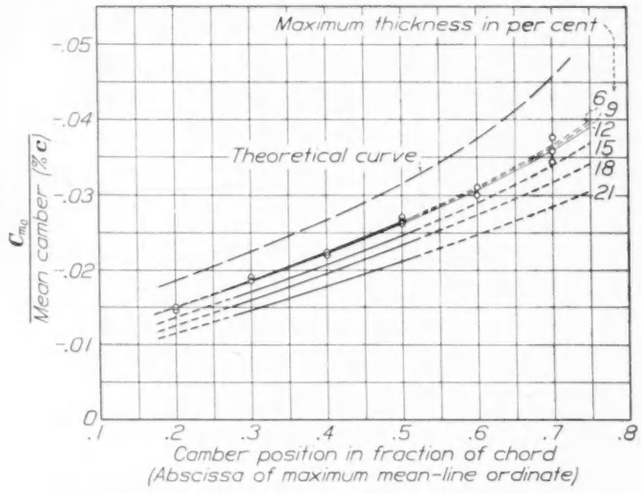


Fig. 19—Variation of moment at zero lift with camber. Points shown are for 12 per cent thick air-foils. Curves indicate general trends for the different thicknesses

moves back along the chord. Referring to Fig. 19, the plotted data indicate that the moment coefficients are nearly proportional to the camber. It will also be noted that the curves representing the experimental values as ratios to the camber are nearly parallel to the equivalent curve representing the theoretical ratios except that the curves tend to diverge for positions of the camber well back. The actual values of the moment coefficient, however, are considerably smaller numerically than the theoretical values.

In concluding, it should be pointed out that representative data have been given applicable to full-scale airfoils of approximately conventional form. In addition, information has been given on which to base an estimate of the effects of variations of scale and turbulence. The subject of the aerodynamics of wing section is, of course, not disposed of by any such treatment; we need to know more about other airfoil shapes, particularly other leading-edge forms, and the effects of adding auxiliary devices, such as flaps, slots, ailerons, spoilers, auxiliary airfoils, etc. The information given, however, establishes a foundation on which to build. Wings with auxiliary devices almost always require the use of efficient basic airfoil sections.

In regard to the application of the data by designing engineers to the problem of selecting the most suitable airfoil section for a given purpose, we believe that most practical engineers will apply the comparable data for different sections to trial designs carrying each design far enough to determine the effect on the over-all characteristics of systematic variations of the airfoil-section shape.

Aims and Methods Changing in Tire Noise Study

PUBLIC taste has changed in respect to car noise.

Twenty years ago the use of muffler cut-outs was nearly universal, and the thrill of passing someone was incomplete unless your motor had a louder, deeper roar than his. Gradually, noise which had given added exhilaration to the early motorists became a general nuisance. As the car mechanism was improved, noise was reduced, until, within the last few years, tire noise has become noticeable, and its reduction has been required, especially at speeds below 40 m.p.h. In most cars, wind noises are sufficient to mask tire noises above this speed. However, as streamlining becomes more complete, wind noises will be further reduced, and tire noise at higher speeds will become more important.

Tire noise is objectionable, not only because noise in itself is disagreeable, but also because the owner quite often mistakes tire noise for axle noise, timing-chain noise, or other mechanical noise, and demands adjustment of these parts when there is nothing wrong with the car.

However, while tire noise has been very generally objectionable to car engineers, there are still some car owners who like to hear certain types of tire noise, especially a low hum which gradually rises in pitch as speed is increased. To one with a musical ear, such a pitched sound serves as a simple and quite accurate speedometer for maintaining constant speed.

The first difficulty in the study of tire noise is that of suitable methods of measurement. Audiometers or acoustometers of various types are available, and they have been used for the measurement of tire noise, or of combined car and tire noise. While it is possible to analyze complex sound mixtures, and to plot a sound curve showing the intensity of sound at different frequencies, it is difficult to interpret such a curve in terms of practical car experience, or to determine which frequencies are troublesome, and which are harmless.

Tire noise at constant speed may be largely a fundamental note with a single frequency, or it may be a very complex mixture of tones and noise with a wide range of frequency. While total noise within any desired range of frequencies may be measured and expressed in decibels, it is doubtful whether this is of real value, since the character of the noise is fully as important as its total volume.

The ear is more sensitive to vibrations of certain frequencies than to those of other frequencies, and unless this is considered in audiometer measurements, the results are likely to be in error.

Because of these and other objections which can be raised against the audiometer methods of determining tire noise, we have preferred to use direct tire-to-tire comparisons on cars under actual operating conditions, or on test wheels sufficiently close to operating conditions to give reliable results. After a little experience, it is possible to make comparative tire-noise tests on cars on a range of road surfaces with quite reproducible results, if direct comparisons are made within a short interval. However, it is difficult to remember sound intensities and characteristics, so whenever it is necessary to test a new type of tire, it is advisable to make direct comparisons with a previous standard.

Still more accurate comparisons can be made by running the tires alternately on a large test wheel. In this way two or more tires may be readily compared and the observers can listen repeatedly to the noise from each until very definite conclusions can be drawn as to the characteristics of the noise. Phonograph records may also be made under these conditions to serve as permanent records of the sound.

The howling or squealing type of noise produced when cars are driven rapidly around turns can be studied most satisfactorily by special types of road tests.

—Excerpt from paper at 1934 Annual Meeting, Detroit, by Arthur W. Bull, Development Division, U. S. Tire Co., Inc.

Effects of Front-Wheel Stability on Public Safety

By Walter A. Olen
President, Four Wheel Drive Auto Co.

AFTER submitting evidence to show the number and character of highway accidents involving motor vehicles, Mr. Olen points out that there are many factors that affect the front-axle stability of a vehicle and goes on to discuss the relation of these factors to public safety on the highways.

He presents the results of tests comparing the effort required on a four-wheel-drive and a two-wheel-drive truck to hold the steering wheel at a given angle when turning circles on a smooth, dry pavement, as well as test data bearing on various other phases of front-axle stability.

NEED for diligent study and prompt action to reduce the loss of life and property as a result of the operation of motor vehicles on our streets and highways is paramount. Use of trucks upon the public highway is still in a very unsettled condition. Railroads and other types of transportation are using their influence to minimize the importance of the use of commercial trucks on the highways. Bankers and investors are favoring the railroads in which their money is invested as against highways that are Government owned and controlled. Legislators are concerned because approximately 30,000 people are killed, 1,000,000 injured, and approximately $2,000,000,000 worth of property damaged or destroyed upon the highways each year. The highway departments of several states are concerned about the kind of roads to build in the future and the maintenance of roads as now constructed.

There are approximately 24,000,000 motor vehicles operating on our highways. One corporation has an investment of over $30,000,000 in truck equipment alone, representing 80 per cent of its capital assets. It is and should be concerned about the future of trucks as a dependable means of transportation.

[This paper was read at the Annual Meeting of the Society, Detroit, January, 1934.]

Road building and the manufacturing of trucks and automobiles have not been coordinated and each has been developing more or less independent of the needs of each other in the interest of public safety. Uniting the automobile and truck manufacturers with the users and allied industries would perfect an organization powerful enough to meet the railroad competition, and also aid materially the American Road Builders' Association, the Federal, State, County and Municipal road and street building and maintenance in securing public funds to complete the roads system of America and also to improve roads that materially reduce road accidents of all kinds.

To determine the trend of thought, clippings of newspapers in America were gathered by the Four Wheel Drive Auto Co. since 1931 on truck accidents. There has been a steady increase in clippings from month to month. In October 1933, our organization alone gathered 2807 articles in newspapers on truck accidents. The circulation of these articles amounted to 134,309,178. Thus this question of truck accidents is constantly and vividly before the people.

We have also gathered editorials to learn the attitude of the press itself and in November, 99 editorials appeared and these have been growing more bold and aggressive, such as,

"Who Owns the Highway"
"HighwayWreckers"
"Bus and Truck Chaos"
"Highway Peril"
"Road Hog Trucks"
"The Truck Hazard"
"Streets for Freight Yards"

In the State of Connecticut during the year 1932, a total of 14,316 accidents was reported, with 402 killed, 12,276 injured—a personal injury to nearly every accident. In the United States as a whole the average is higher, there being 5 persons injured to every 4 accidents. In Connecticut there is a total of 350,716 registrations. Of the total accidents, 7246 were reported as commercial. Classified, these accidents show that 699 were head-on, 1925 rear end, 2280 side-swiping, 154 backing, and 2188 front against side.

All the other accidents were collisions with pedestrians, other vehicles or objects, collisions with stationary objects,

and no collision accidents. Under the latter item appeared overturning, of which there were 324, and fire 29, the only accidents to show an increase during the year.

Head-on, side-swiping, front against side and overturning caused 7416 accidents, or, out of a total of 14,316—52 per cent, were of a nature that might be affected by the stability of the front axle.

Regulation is strict in Connecticut. Connecticut is the only state where at least 20 per cent of the gross load must be on the front axle. As this law has been in effect for the last five years, we can assume it is reasonably observed. Educational work has been done by the state. Accidents are reported separately and are not arrived at by the rate of averages; responsibility for each accident is fixed at the time of the accident and is classified. The public mind is safety conscious and the general reduction in accidents shows that its educational work is effective. Connecticut has a very large percentage of hard surfaced roads. It has done probably more to sand and otherwise care for wet and slippery roads than any other state. Fifty-one per cent of all the accidents are truck accidents. In the United States, trucks represent 13.5 per cent of all registrations, but are responsible for 16 per cent of all accidents. Out of the 7246 commercial vehicle accidents in Connecticut, 2280 were caused by side-swiping, 2188 resulted from collisions of front against side, making a total of 60 per cent, or, 4468 accidents caused in this manner. The accident record of this state indicates clearly that even 20 per cent of the gross weight on the front axle is not sufficient to control the direction of the vehicle under all varying operating conditions.

Accidents Analyzed

Generally, road conditions are considered responsible for only 5 per cent of the total accidents, but snow, rain, and fog are charged with 15 per cent. It has been shown that most motor-vehicle accidents occur under ideal operating conditions. Statistics show that:

91.71 per cent of all accidents were by men, not women.

74.84 per cent of all accidents were by men from 25 to 64 years, not reckless youths.

94.23 per cent of all accidents, the car or truck was in good condition.

82.18 per cent of all cars in accidents had brakes on four wheels.

88 per cent of all accidents were on a straightaway road.

47 per cent of all fatal accidents in Wisconsin were on 7 per cent of the hard surfaced roads, 5600 miles.

53 per cent were on the remaining 78,000 miles of secondary roads.

70 per cent of all accidents in the United States were at speeds less than 30 m.p.h.

4 per cent were at 10 m.p.h. and only 2 per cent at 50 miles or over.

Recent legislation has limited the load which may be placed on an axle. The present trend in motor-truck design is, therefore, to place more load on the front axle, to enable attaining a greater gross load and payload. But how many manufacturers know how far they can go in this direction without decreasing safety by decreasing the maneuverability of the vehicle? Some authorities claim 9000 lb. the maximum without power steering—others go as high as 10,000 lb. But does power steering rectify any possible inherent defects in the design of the axle and steering mechanism? Does it enable the vehicle to follow the turned front wheels any more readily? Tests we have conducted show that a truck with power driving, the front as well as the rear wheels may carry 14.2 per cent more load on the front axle than when operated as a rear-drive truck, with the same steering effort.

FWD trucks have been successfully operated with front axle loads as high as 13,000 lb., without power articulation, and with no difficulty encountered in steering.

Weight Distribution

How much of the total weight to be propelled *should be* on the front or guiding wheels? A recent questionnaire sent out to truck manufacturers revealed a wide variety of opinions as to the minimum per cent of the gross load that should be carried on the front axle of a motor vehicle. The following were recommended:

No. 1 Manufacturing both automobiles and trucks 10 per cent.
No. 2 Manufacturing both automobiles and trucks 18 per cent.
No. 3 Manufacturing both automobiles and trucks 20 per cent.
No. 4 Manufacturing trucks 21 per cent.
No. 5 Manufacturing trucks 22 per cent.

Two manufacturers gave the following as standard ratings for new models:

No. 6 Manufacturing both automobiles and trucks 20 per cent.
No. 7 Manufacturing trucks 30 per cent.

Many trucks we checked showed as low as 5 per cent.

Commercial Car Journal gives the general practice 10 per cent.

One large fleet user claimed his fleet varied 7 to 13 per cent.

P. M. Heldt, in his text book, gives the ideal weight at 33 1/3 per cent.

Front-axle stability in the interest of public safety is an interesting subject, but one concerning which it is very difficult to secure any information outside of mere mathematical calculations and mental conjecture. There are many factors that affect the front-axle stability of a vehicle, such as load distribution, variation of coefficient of friction of the road surfaces, acceleration and deceleration, crowned roads, side winds, grades, curves, powered and non-powered front axles, tires, springs, etc.

Tests conducted by J. Bradley, B.A., and S. A. Wood, M.Sc., and published by the Institution of Automobile Engineers, London, England, showed that when the center of gravity is approximately central, front-wheel locking exerts a greater retarding force than rear-wheel locking. In addition, a straight path is followed with locked front wheels as against an extremely uncertain path with rear brakes alone. With four-wheel brakes the location of the center of gravity did not change results, while on all other brake applications it made material differences. It was further found that the location of the center of gravity did not influence either distance to stop or the direction. The present height of trucks, now almost uniform, may therefore be not a material factor so far as braking is concerned, with efficient brakes on all four wheels.

With brakes on only the rear wheels, erratic braking conditions are encountered, the steerability of the vehicle varying

with the static weight distribution. The usefulness of two-wheel braking systems will also vary with the height of the center of gravity, as the latter determines for a given wheelbase the load shift to the front axle and thus the retarding force when the two wheels are locked.

Braking Effects

It is a recognized fact that a greater braking effect results when the wheels are *almost* locked than when the wheels are *locked and sliding*. However, with brakes in perfect adjustment, set to *almost* lock the wheels, application of the brakes on a crowned road or on a turn may, due to the transverse force, cause the wheels to lock and slide with entire loss of steering effect. It is apparent that should both front or both rear wheels lose their traction and lock and skid, that end which loses traction may move sidewise, due to the transverse force, placing itself in the way of other traffic, or moving toward a permanent obstruction at the side of the road, thus promoting an accident. Control of the vehicle can be regained only by releasing the brakes completely, restoring "steerability," and then re-applying them more gradually. This is an actual operating condition that occurs frequently and suggests that under many road conditions, vehicles have brakes that are too positive.

When a vehicle is stopped by the application of brakes, a rotative effect is produced, which increases the load on the front wheels and decreases that on the rear. This shift in weight, with wheels locked and sliding, may easily be 20 per cent of the gross vehicle weight. (It will be, with the height of the center of gravity, one-third of the wheelbase, and the coefficient of friction 60 per cent.) Thus if a truck is designed to carry 50 per cent of the total weight on the front axle, the weight will increase to 70 per cent during a sliding stop, or a load 40 per cent greater than the axle was designed to carry. On the other hand, a truck designed for 20 to 80 per cent weight distribution, will have 40 per cent of the gross weight on the front axle during such a stop, or an overload of 100 per cent on the front axle. This perhaps explains why some trucks have strained front axles and require excessive attention to their front wheel brakes and tires. It certainly does explain why the conversion of passenger cars into trucks has often failed.

Brakes and Load

It appears, in general, that a static loading in front of 33 per cent to 40 per cent will give the best all around braking performance, as the load will be distributed almost evenly front and rear during sliding stops.

In a similar manner weight is shifted from the front axle to the rear during periods of acceleration. The gear ratios of most vehicles are so proportioned that an acceleration of at least 15 ft. per sec. per sec. can be obtained in low gear on a level roadway. If a rear-drive truck be loaded to 85 per cent on the rear and only 15 per cent on the front, an acceleration of only 14 ft. per sec. per sec. will shift all the weight from the front to the rear. Any increase in acceleration will cause the front wheels to be lifted from the ground, with obvious effect on the steering. A vehicle under these conditions, should one of the rear brakes be dragging slightly, will go through unexpected gyrations and may cause an accident on the highway.

Another point which affects the safety of a vehicle and which should be taken into consideration in front-axle design is the effect of bouncing on steerability. Upon ascending an obstruction or even a slight wave in the road surface, the front end of the vehicle may receive a sufficiently great upward acceleration to leave the road temporarily. The greater the speed, the greater this tendency toward loss of control, which may result in not getting out of the way of traffic in the other lane. It can be shown that the velocity with which a vehicle may safely pass over an obstruction in the road is greater with a greater load on the front axle.

It is obvious that the front axle of a truck with its guiding wheels is very important in determining the direction of the truck under the varying road conditions and at the different speeds. Here again we are indebted to English engineers for some interesting figures. In view of the fact that over half of all accidents are on 7 per cent of the roads in America that are hard-surfaced and that 88 per cent are on the straight-away and 70 per cent at speeds less than 30 m.p.h., it is well to inquire into the side force coefficient variation on various surfaces.

	Dry	Per Cent Variation	Wet	Per Cent Variation	Per Cent Variation Wet and Dry
Waterbound Macadam	0.87	None	0.71	None	20.2
Tarred Macadam	0.85 to 1.09	24.7	0.80 to 0.97	19.3	30.6
Wood Blocks	...	...	0.57 to 0.92	47.0	...
Asphalt	0.74 to 1.05	34.6	0.71 to 0.98	32.0	48.6
Concrete	0.88 to 1.04	16.7	0.54 to 0.88	48.0	63.0
Mastic Asphalt	0.67 to 0.76	12.5	0.61 to 0.80	27.0	27.0
Asphaltic Concrete	0.99 to 1.02	3.0	0.90 to 0.93	3.3	12.5

Tests recently made in Ohio corroborate the above differences, in part, at least.

Differing Road Surfaces

A driver who safely directs his vehicle from one form of road surface condition to another must possess a great amount of judgment and skill, or else there must be provided a substantial margin of safety in the design of the front axle and load to allow for the variations if safety is to be considered of paramount importance.

The large variation in road conditions between rough and smooth dry, the large variation between the same road in a wet condition and the still larger variation between wet and dry, together with this shifting load, offers a possible explanation to why so many more accidents occur on the straight-away road.

Crowned roads present another highway danger. A vehicle coming to a sliding stop on a crowned road will drift to the right a distance of more than a foot. The amount of drift is hardly perceptible to the driver, and he is often given no warning until the right wheels drop off the pavement, or encounter softer material, when the steering wheel may be jerked from his hands and turned so abruptly that the vehicle will go into a skid or spin with possibly disastrous results.

The recent advent of low-pressure tires and the present tendency toward softer front springs create a condition which may seriously affect stability. Centrifugal force throws a large part of the load from the inside wheels to the outside wheels in rounding curves. The outside wheels may be carrying as much as 60 to 70 per cent of the total weight. The resulting tendency toward tilting should be carefully considered in the design of the front axle, springs, and steering mechanism.

Another factor affecting front-end stability is the pressure exerted on the side of a vehicle by a side wind. While this is often neglected, it may be shown that a 10,000 lb. truck

with 120 sq. ft. side area may be moved bodily sidewise by a 35-40 m.p.h. wind on an icy road. Average hourly wind velocities, tabulated for over 50 of our largest cities where constant records are kept, show monthly averages as high as 17.8 m.p.h. In New York City alone, according to the U. S. Weather Bureau, a wind velocity of 32 m.p.h. or more was recorded on 101 days during 1932.

Under all these varying conditions power within the guiding wheels enables them to respond more readily to the deviated direction and the shifting load is timely so as to contribute to greater traction, thus enabling it to direct the load in the direction desired.

It has been demonstrated by means of a hydraulic drag link that a truck steers more easily with power on all four wheels than with power on the rear wheels only. The following test was conducted by comparing the effort required to hold the steering wheel at a given angle when turning in circles on a smooth, dry pavement. The first test was conducted with a standard FWD 2½-ton truck; the second with the same truck used as a rear-drive by disconnecting the drive to the front axle. The truck was loaded to 5680 lb. on the front axle and 8520 lb. on the rear, giving a distribution of 40 to 60 per cent.

Speed in m.p.h.	Angle of Wheel Cramp in Degrees	Lb. Effort at Rim of Steering Wheel	
		4-Wheel Drive	Rear Drive
5	24	28	35
10	24	56	63
15	24	63	70
5	18	14	20
10	18	28	34
15	18	38.5	45
5	12	10.5	14
10	12	21	26
15	12	31.5	37.5
5	9	5	7
10	9	10	14
15	9	18	24.5

Analysis of the above figures shows that the average effort required to steer the four-wheel-drive truck was 14.2 per cent less than that required to steer the same truck operated as a rear-drive truck.

Further tests showed that no more effort was required to steer a truck when a four-wheel trailer or a two-wheel semi-trailer was attached than when the truck was operated alone. These tests were, of course, conducted with the same load on the front axle of the truck.

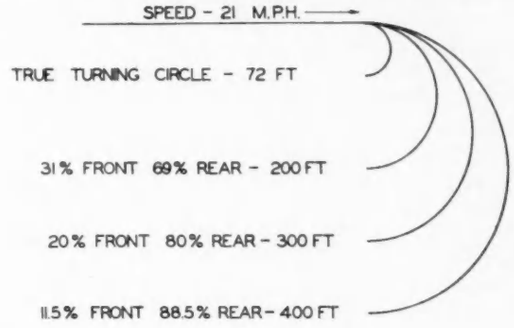
In addition to the advantage of greater ease of steering a truck with power on the guiding axle as well as the rear, there is the fact that the front wheels with power aid materially in changing the direction of travel of the vehicle when they are turned. This is explained by the fact that in a rear-drive truck the driving force is always in line with the axis of the vehicle and, therefore, at an angle to the plane of the front wheels when they are cramped. At such times there is a component force perpendicular to the plane of the wheels, which creates a tendency toward skidding. This component increases rapidly as the turning angle increases. With power on the front wheels, the driving force is always in the direction the wheels are turned, and this minimizes the tendency to skid.

Tests indicate that, on a properly proportioned four-wheel-drive truck, full and free acting differentials in all wheels

Turning Circle Tests Conducted on Smooth Frozen Ground Covered with 4 to 6 In. of Snow

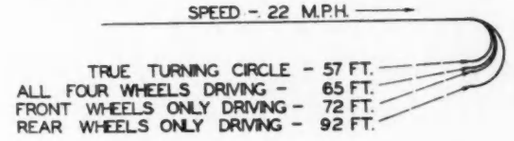
Test of Rear-Drive Truck with Variable Weight Distribution

176 in. wheelbase—24 deg. cramp angle
34-7 tires dual rear—inflated to 95 lb. per sq. in.



Test of F.W.D. Truck

12,300 lb. gross weight—40 per cent front—60 per cent rear
138 in. wheelbase—24 deg. cramp angle
34-7 tires single rear—inflated to 95 lb. per sq. in.


SPEED - 22 M.P.H.
TRUE TURNING CIRCLE - 57 FT.
ALL FOUR WHEELS DRIVING - 65 FT.
FRONT WHEELS ONLY DRIVING - 72 FT.
REAR WHEELS ONLY DRIVING - 92 FT.

when speed is desired is necessary in the interest of public safety because of the constantly changing diameter of wheels with pneumatic tires, the constantly shifting load and the changing road conditions.

A 3½-ton truck was tested with 8600 lb. on the front axle, 13,600 lb. on the rear axle, giving a weight distribution of 38.7 per cent on the front; tires 10.50 x 20 balloons, singles front and rear; with cyclometers connected to each wheel. The distance run was 25.7 miles, of which 8 miles was concrete, the balance gravel. During the first test the roads were wet and the grades slippery. Most of the turns were right hand.

Cyclometer reading, right front wheel, 12,925; left front wheel, 12,935.

Cyclometer reading, right rear wheel, 13,122; left rear wheel, 12,980.

The mean difference in rotation between the wheels on the front and rear axles, according to the cyclometers, was 121 revolutions. Since, due to the action of the axle differential and to the shifting load, slippage would occur on only one wheel at a time, and the rotation of that wheel would be twice that of the ring gear. Therefore, the total wheel slippage, which would have been encountered had there been no free-acting third differential between the two axles, would be twice the above amount, or 242 revolutions, corresponding to a slippage of 2495 ft.

A second test was run with the same truck, load, etc., but

road conditions were fairly dry. The general direction of the turns was changed, most of them being toward the left.

Cyclometer reading, right front wheel, 12,945; left front wheel, 12,960.

Cyclometer reading, right rear wheel 13,060; left rear wheel, 12,975.

In this test the difference in rotation of front and rear axles shows a difference in the revolutions of the wheels corresponding to a slippage of 1340 ft.

These tests show that there is always a difference in rotation of the wheels of the front and rear axles. On the curves the front wheels rotate more, due to their turning in a larger circle; on the straightaways the rear wheels will rotate more if their load per tire is greater than that of the front wheels.

Without free-acting differentials the difference in travel would have to be taken care of in wheel slippage and wheel slippage produces an internal strain on the driving mechanism. Also the shifting load on driving wheels increases the steering pressure and the wheel slippage causes a momentary loss of control. It further causes the guiding wheels to drift until the strain is released.

Allowance must also be made for over-steering and under-steering. The crown of the road, changing center of gravity and high winds are real factors in accounting for the large number of accidents on the hard surfaced straightaway. Examining the accident records of a road between Milwaukee and Chicago show three times as many rigs going off the east side of the road than the west side because of the prevailing west wind.

Also there must be considered the various grades, curves, oil, dirt, clay, dust with and without moisture, ice, snow, sleet and other variable conditions together with the greater variation on soft or secondary roads.

In addition to the danger of trucks without sufficient load on the front axle is the evil of overloading, which is a substantial contributor to fatal accidents. The Department of Transportation Coordination at Washington, D. C., after computing costs of operation on over 13,000 fleets of trucks, states most definitely that there is no economy in overloading. The cost will be the same whether you have 2 tons on a 2-ton truck, or 4 tons on a 2-ton truck, but the liability for accidents has been increased and the dependability and reliability of the vehicle reduced.

This paper has been prepared to show the importance and need for further study on this subject. It is hoped that those interested in increasing the safety on our highways will be encouraged to further study the many variable factors that contribute to front-axle stability.

Discussion of Olen Paper

Thinks Load Shift Effects Overstressed

—B. B. Bachman
Vice President in Charge of Engineering, Autocar Co.

COMMENTS made by Mr. Horine on the pertinence of statistical information given in Mr. Olen's paper are supported by the figures.

The question of the effect of load shift produced by deceleration of the vehicle, braking, and front axle design, would seem to be somewhat overstressed.

It is true that the order of increase in weight, which Mr. Olen refers to, occurs in practice but it is also true that, when the vehicle is passing over bumps in the road, it has been proven by investigation through the Bureau of Public Roads that the increases in load on axles, etc., are approximately of the same order. These conditions must be recognized and allowed for in any proper design of vehicle. The comments made as to this factor being one of the contributing causes to the dissatisfaction experienced with the attempt to convert passenger cars into trucks is, however, quite to the point.

The statement is made that the load shifts to the outer wheels when rounding a corner, due to centrifugal forces increasing the loading on the outer wheels by an appreciable amount. This is also recognized, but the question of the interjection of softer springs does not seem to be pertinent to the subject.

While it is recognized that in a rear-driven vehicle the driving forces fly outside of the path of the vehicle when rounding a turn, it is also true that this condition exists to some extent in a four-wheel-drive vehicle for the reason that very generally the rear driving forces are in excess in the front-wheel forces and will leave the condition still present to some degree.

There is no question that the safety of vehicles can be improved by a comprehensive and detailed study of the items to which Mr. Olen refers, as well as by study of many others which have not been included in this paper. All of these items deserve careful consideration under present day operating conditions.

Believes Statistics Wrongly Interpreted

—M. C. Horine
Sales Promotion Manager, International Motor Co.

IT would appear from Mr. Olen's statistical presentation, that the motor industry had learned from the railroads some of the uses of statistics, which brings to mind the startling fact as calculated by a statistician that if a fox terrier 2 ft. long from tip to tip, with a tail 1½ in. long could dig a hole 3 ft. deep in 10 min.—then to dig the Panama Canal in one year it would require one fox terrier 15 miles long, with a tail 1½ miles high.

As I review Mr. Olen's statistics regarding accidents to different types of vehicles, they appear to prove the exact opposite of what he claims inasmuch as most of the accidents occurred to passenger cars, which, as is well known, carry approximately 50 per cent of the weight on the front axle.

However, I feel that a weak argument in support of a

proposition does not necessarily invalidate that proposition and I do not think that these accident statistics are necessary to prove Mr. Olen's contention, namely, that more equitable distribution of weight as between front and rear reduces skidding and increases the positiveness of steering, for with the type of buses which now prevail in our cities, wherein the distribution of weight is more nearly equal, front and rear, it has certainly been found that skidding is almost impossible.

Differs with Author on Analysis of Data

—A. W. Herrington
President, Marmon-Herrington Co., Inc.

I HAVE gone very carefully through the statistical analysis that Mr. Olen has made in this paper and there are some instances in which I would like to suggest an interpretation of these statistics, entirely different than the one chosen by the author. There are some cases, where, in my opinion, the statistics cited do not support the conclusions which have been drawn from them.

As an engineer I have always found that in the employment of research associates, it is an extremely difficult thing to secure men who possess a sufficiently open mind, to permit the results which are obtained from the experiment to be intelligently used.

Engineering tests and research work can be conducted in two ways: The first is to conduct an experiment for the purpose of proving a statement which is assumed to be true before the test is conducted; the second is to secure the best possible average result from the test, taking into consideration all of the variables which may have a bearing, and then weighing the final data for such truth as may be gleaned from them.

In the general consideration of the problem of placing safe cargo-carrying motor-vehicles on our highways, a joint responsibility exists between the producers of these vehicles and the highway control officials of the various states in which they are used. I believe I can positively state that there is no producer of trucks in the United States today, who would knowingly sell or deliver a vehicle which would be inherently unsafe in operation. The powers of discernment which the education of experience has developed in our motor purchases, would insure the rapid business suicide of any concern attempting to do such a thing. There is one very important fact in this connection which I believe the author should have stressed. It is possible for any company to deliver a perfectly safe and sound motor-vehicle to the user and for the user so to equip and load this vehicle that it is inherently unsafe from an operating standpoint and becomes a danger to all other users of the highway. It is at this point that the traffic regulating authorities come into the situation. If more attention were given to the safe equipping, and safe loading of vehicles of the cargo-carrying type, the percentage of accidents which occur with these vehicles would be materially reduced.

I do not mean when I make these statements that the operator knowingly causes a motor-vehicle to become a dangerous instrument upon our highways. When these things occur they are usually due to the lack of technical knowledge pertaining to the disposition of weights by the users. It is at this point that the technically trained and experienced experts of our State automotive regulating bodies should step in to assist and to educate those operators who are not equipped with the proper knowledge to help themselves.

In this joint responsibility between the automotive industry and the state regulating authorities there exists a very grave danger to our industry. This is the thing we must always carefully guard against. Such education as we are able to bring to bear upon the situation should always have the effect of keeping in the hands of the regulating authorities full power to control the proper loading and equipment of motor-vehicles, but of avoiding formulation of regulations which might throttle the industry by crossing the borderline into the field of automotive design.

I now desire to comment at random on some of the conclusions which Mr. Olen has seen fit to draw.

He refers to 7416 accidents classified by the State of Connecticut, as head-on, side-swiping, front against side, and overturning, and makes the statement that these were of a nature that they might be affected by the stability of the front axle. It would be appreciated very much if he would enlighten us on the subject of how a head-on collision or a collision of the front of a vehicle against the side of another one, could be in any way influenced by front-axle design. Possibly the physical limits of the paper have been such that he has not been permitted to fully expand his thoughts along this line.

I would like to suggest that in weighing the accident record of any particular community that it is necessary also to take into consideration the traffic density flowing upon the highways of that community. The use of percentages in statistics is a thing which may often be misleading.

This question of weight on the front axle of the vehicle is one of extreme interest to all designers at this time. The practice of the highway authorities in limiting individual axle loads has had a tendency to cause designers to increase the relatively light load which the author has indicated has been carried up to this time on the steering axle of the truck.

Comment on Tests

Mr. Olen describes a test with a 2½-ton four-wheel-drive truck on dry pavement with a front-axle loading of 5680 lb., and he has reported the pounds of effort at the rim of the steering wheel necessary to accomplish the desired steering result. Then by disconnecting the front-axle driveshaft he has made a similar analysis, but by using a four-wheel-drive truck with its heavy front-axle driving and steering units as being representative of a rear-driven truck. I question the conclusion which he has drawn from these results. First, the truck used to conduct the test could not be fairly considered as giving results which are comparable with the average rear-driven 2½-ton trucks which are being delivered to the customers today. The very nature of the four-wheel-drive design makes it impossible to secure any analysis on this basis which could bring about intelligent results.

I would further suggest that the steering efforts recorded by Mr. Olen's engineers for this 2½-ton truck are about 300 per cent more than the present existing normal practice of the truck industry. This statement is intended to include both standard rear drive and available standard four-wheel-drive trucks.

It would be interesting to learn why these observations were limited to a front-wheel cramp of 24 deg. Why were they

not continued to include the normal maximum cramp of 30 deg. on the inside front wheels?

Experience has shown us that the largest individual factor influencing the effort which it is necessary to exert on the rim of the steering wheel is the caster angle of the front axle. In a four-wheel-drive truck, where the power is transmitted through the Hotchkiss-drive type of front-spring installation, it is necessary to make allowance for this torque reaction in determining the caster angle to be used. To obtain the same steering effects, the caster angle of a four-wheel-drive truck must be set at less than common rear-drive truck practice, when the vehicle is standing still. When power is applied, torque reaction increases the static caster angle, the amount being determined by the stiffness of the front springs. For this reason alone, comparable results on the basis chosen by Mr. Olen are probably unfortunate in not representing the true picture.

Location of Center of Gravity

Mr. Olen has, in my opinion, touched rather too lightly on the subject of the effects of the location of the center of gravity. It is true, the height of motor-trucks is controlled by existing regulations, but within this limited height will be found a very wide variation in the height of the center of gravity of trucks in use today.

I want to take direct issue with the statement that the location of the center of gravity has no effect on the braking performance of the motor-trucks.

As a homely illustration, may I suggest that a pair of roller skates having relatively short wheelbase, attached to the feet of an individual like myself, having relatively high center of gravity and no experience with the use of these appendages, would probably demonstrate very quickly that there is some relationship between center of gravity, wheelbase, and general stability.

The height of the center of gravity in relation to tread and whether single or dual rear tires are fitted, has a very definite effect on vehicle lateral stability against side overturning forces.

The author seems to later concur in this opinion, because further on in his paper he refers to the effect of the height of the center of gravity on the transfer of weight on vehicle axles. Under "Braking Conditions," the amount of weight transfer from axle to axle is very largely governed by the height of the center of gravity with reference to the wheelbase.

Special Cases

In justice to Mr. Bradley and Mr. Wood and their paper to the Institute of Automobile Engineers, it would be suggested that the statement which they made with reference to the effect of the center of gravity on the stopping of motor-vehicles was intended to be confined to the particular series of vehicles which they were using for their experiment at that particular time, and that they did not intend to use this statement as applying to all vehicles on the highway.

Probably one reason why the height of the center of gravity has not caused more difficulty in the case of motor-trucks, is that it would be an extremely difficult thing to conceive of a design which would be capable of either deceleration at 15 ft. per sec. per sec. when fully loaded, or of acceleration at this same rate, as suggested by Mr. Olen in his analysis. In fact, to produce a vehicle which would be capable of such a violent acceleration, even in low gear, would require an engine of a horsepower considerably larger than the average sizes which have been found to be economically usable by industry.

I took the opportunity to discuss this question of acceleration with the American Automobile Association officials and also to seek some records with reference to a certain 16-cylinder motor car which has been very completely performance-tested on the Indianapolis Motor Speedway. This vehicle developed 1 b. hp. for every 28.2 lb. of gross loaded weight, under the conditions under which these tests were held. The high gear acceleration with 3.78 axle ratio was 4.7 ft. per sec. per sec.; in low gear with 11.6 overall ratio it was 14½ ft. per sec. per sec., but in this start the rear wheels were spinning on the dry brick surface immediately after the clutch was fully engaged. For this reason I question the possibility of any usable truck design which could possibly have such a violent acceleration rate as that indicated by Mr. Olen.

Mr. Olen states that "in any four-wheel-drive truck full and free-acting differentials effective upon all wheels are necessary in the interest of public safety." I do not believe he intended to convey the perfectly obvious corollary to this statement which would be that all trucks of the four-wheel-drive type not made with full and free-operating differentials are unsafe and their use inimical to the public safety.

Specific Design Practices

To show the extent to which Mr. Olen's statement may be questionable it will be necessary to review the existing situation in the industry and to cite some specific trucks by name. The Walters truck is a four-wheel-drive truck having three differentials. The designer of this vehicle, however, seems to disagree with Mr. Olen's engineers and has found it advisable to use three automatically-locking differentials in his design. The reason for this is, that unless something of this character is used, the loss of power to the wheel having the least traction would cause the complete stoppage of the vehicle, if one single wheel were to lose traction.

All models of the Oshkosh four-wheel-drive truck formerly had three differentials. In this case the center differential only was of the locking type. In several new models which Oshkosh has introduced during the past year, they have for some reason found it expedient to eliminate this center differential and to use an auxiliary transmission which does not provide any differentiating mechanism between the front and the rear axles.

Coleman Motors Corp., the White-Indiana organization, and my own company, all make four-wheel-drive vehicles which have no inter-axle differential. We also use axle differentials of the high traction type.

It is my opinion that this question is purely of academic merit. It certainly has no more weight than a similar discussion upon inter-axle differentials on rear-driving two-axle bogies. It most definitely has no bearing upon whether a vehicle is, or is not, safe in the hands of the using public.

During this week at Washington, D. C., the U. S. Army is accepting bids upon a large quantity of all-wheel-drive motor vehicles of all types. The specifications upon which these proposals are based clearly state that no center differential will be required, but that if it is supplied, that the company choosing to furnish it must provide some clutch means of eliminating its action for cross-country service. We further note that the trucks produced by Mr. Olen's company are all fitted with this center-differential lock-out clutch mechanism.

The inter-axle differential on a four-wheel-drive truck is essentially a power dividing mechanism. It will always subdivide the power equally to the front and rear axles. This fact probably accounts for the unusual front wheel tire wear that is always observable on trucks possessing center differentials of the "full and free-acting" type. When these vehicles get into a heavy pull, the front axle carrying the lightest load, the wheels spin on the pavement causing undue tire wear.

It is further noted out of the actual operating records of users, that there is no greater front-tire wear on vehicles possessing no center differential than would be expected upon a truck of the standard rear-driven type.

Theory and Practice

This discussion now enters that realm which has been aptly referred to by C. F. Kettering in the following manner: "When theory and practice do not agree, you should examine the facts to see what is wrong with the theory."

The facts are that the tread of the front axle on most of the four-wheel-drive trucks which do not possess a center differential approximately lines up with the center of the two dual wheels fitted at the rear. In turning a circle, the track described by the outer front wheel will be approximately the same distance outside of the outside dual rear wheel as the tracks of the two duals are from each other. It is quite obvious, in turning a corner with an ordinary dual installation where the wheels are solidly bolted together, that there must be some tread slippage. The amount of tread slippage that occurs between the front wheel and the rear wheels of a four-wheel-drive truck not having a center differential is no greater than the slippage between two duals, if the above stated design conditions are adhered to. This being true, and taking into consideration the adequate and satisfactory mileage which is being obtained from thousands of rear dual installations throughout the United States, it is apparent that no greater front-tire wear could be expected on a four-wheel-drive truck with no center differential than would be normal to a dual installation.

Elimination of front wheel tire wear on vehicles of this type is caused by the fact that the lack of a center differential prevents the overrun or spinning of the front tires when they are carrying the lesser load.

Mr. Olen has outlined a test conducted to prove that the rear axle of a four-wheel-drive truck will overrun the front axle of the same truck in normal operation.

Differing Conclusions

The figures I will use to determine an entirely different viewpoint of this situation are those cited in his paper. He states that for the purposes of the test they used a 3½-ton truck fitted with 20 x 10.50 balloons, single front and single rear. This truck was loaded 8600 lb. on the front axle and 13,600 lb. on the rear.

The normal load which the tire companies recommend and permit upon a single 20 x 10.50 tire is 4700 lb. The two wheels of the rear of this truck would be capable of sustaining a load of 9400 lb. without overloading the tires. For the purpose of this test, however, the author's engineers have apparently loaded this vehicle 13,600 lb. on the rear, or a 4200 lb. overload. This is approximately 44.6 per cent more load than the tire engineers state this tire is capable of carrying. Assuming that the four tires were pumped up to the normal pressure of 75 lb. prescribed for a fully loaded tire of this size, it must be perfectly obvious that the loaded radius of these overloaded rear tires was considerably less than the loaded radius of the more normally loaded front tires. It would be quite normal to expect in a situation of this kind that the rear axle would turn more revolutions than the front axle in order to travel a given distance over the highway.

If the author will refer this question to the chief engineers of any of the tire companies and will secure from them the known loaded radius of a 20 x 10.50 tire when overloaded 45 per cent and will then calculate the difference in the number of revolutions that the rear might be expected to turn more than the front, he will find that his engineers have been fairly accurate in recording the results that they have given to him. The test was completed on one day under rainy conditions and another day under dry conditions. The difference of 500 ft. between the dry day and wet day, I would like to suggest may be attributed to the use of the center differential which permitted the front wheels to slip on up-grade pulls.

Agrees with Thesis

In closing I want to say that I am entirely in accord with the thesis that Mr. Olen attempts to prove. My only regret is that his choice of data and statements may lay him open to the criticism of some of my friends of the rear-drive truck industry and it is for that reason that I have hastened to take issue on these points before our four-wheel-drive industry is criticised by somebody from outside of our ranks.

It has been definitely proven that a satisfactory stable, dependable, safe truck, driven on all wheels can be manufactured and sold.

This vehicle will be found to possess elements of front-axle geometry which are in complete accord with the practices of the rear-drive truck industry of today. After all, the factors which determine a satisfactory front-end geometry are based upon running experience and the developments of many engineers and there is no place where it could be more aptly said that "fifty million users can't be wrong."

The single point which in my opinion determines the stability or instability of a truck which has a steering and driving front-axle is the means of driving the wheels. It has been common practice in the industry up until recently to use the Carden, or pin type, universal joint for this purpose. It is also commonly known that the fluctuating velocity of the driven end or outboard driven shaft of a Carden joint when it is turned to an angle of 30 deg. will be approximately 35 per cent. If there is any single factor which could create front-steering instability in a motor-vehicle, it is due to this particular cause. In addition to this, it does not appear to be necessary in any front-axle design to have the wheel bearing point upon the ground, 4¾ in. outboard of the subtended steering pivot center. The combination of this peculiar conception of front-axle layout with the Carden-type joint makes a reaction through the steering gear which is so pronounced as to be, in my opinion, definitely objectionable. The U. S. Army authorities seem to agree with my observation. Their specifications hold this dimension to a minimum of 1½ in. If a steering gear which is sufficiently unreversible is used, failures of the gear will most certainly result. If the gear is moderately or completely reversible, the resulting gyrations of the steering wheel in any turn of 15 deg. or over, seem to be unsatisfactory from the standpoint of some users.

In view of the fact that Mr. Olen's design has been in production and use for many years I would not have the temerity to suggest that it is dangerous to life and limb to

place such trucks on the highway. In our use of tools and weapons we have progressed materially from the Stone Age and there certainly has been room for progress in the design of front driving and steering axles.

Front-wheel driving joints of the uniform velocity type are available, have been used in the industry, and eliminate all of the unsatisfactory phenomena noted above. Use of such joints combined with the standard practices of normal front-wheel steering geometry will result in the production of an all-wheel-driven vehicle which in every respect will handle as easily and safely as the best motor-vehicles to be found on our highways today.

Value of Free Discussion

If my comments may have appeared to be super-critical, I wish to offer my apology to the author, because in making these statements I have at heart only the best interests of the industry in which we are jointly engaged.

A full and free discussion of any matter that has to do with the safeguard of the public on the highway is not only timely, but I think that Mr. Olen is to be complimented for having possessed the courage to have started such a discussion at this time. Out of a "full and free" comparison of such ideas will come those results which lead to all good things.

Action Is Needed on Accident Study

—**Boris P. Sergayeff**

WHENEVER we try to prove anything by statistics we must be careful not to let our data get confused and then without analyzing them properly derive a wrong conclusion. In this otherwise excellent paper Mr. Olen declares for instance that "91.7 per cent of accidents were by men, not women." No doubt at least 90 per cent of drivers on the road are men and that is why the most accidents fall to men and not to women. Reference to the number of accidents en gros but not in reference to the relative number of drivers leads to an erroneous conclusion. No doubt, however, women drivers are not much better, if not worse than men drivers.

The same can be said about compiling statistics of the number of accidents according to the respective age. Taking an age range between 16 and 85 years, no doubt the number of accidents will be 100 per cent but it will not mean anything. So Mr. Olen's statement that "74.85 per cent of all accidents were by men from 25 to 64 years, not reckless youth", does not prove anything to me, unless I am mistaken.

Laying aside all statistics for the moment it is evident that now is the time to pay attention to the growing number of accidents. Let us look at the roots of driving. We speak so much about reckless driving, but what is a definition of reckless driving and what is careful driving?

First of all a driver is a physiological specimen who can do a certain thing at a certain time. In case of an imminent accident the driver must react in a certain way as fast as his brain can work. To see an approaching trouble, digest it in his brain, take a decision and command his hands or feet to perform a certain movement takes a long time. It becomes, therefore, obvious that drivers must be subjected to a psychological examination in order to find out the speed of reaction of their brains. Therefore to speak about good brakes sometimes is useless because some drivers are so slow in their reaction, so erratic in judgment and indecisive in their command that they should not be allowed even to walk on sidewalks in order not to bump passing pedestrians. This first argument is a key to all accidents.

We are speaking about color blindness, physical fitness and many other things as a requirement for a good driver. We may have rigid physical requirements and go far enough, even actually to find all the items necessary for ideal drivers, but would it not be better to have a standard test under actual driving conditions similar to the test used in England? It consists of a stretch of road with artificial obstructions arranged in a certain system. These obstructions record automatically as soon as they are hit. Scores can be figured as in a golf course with a certain par value. The driver subjected to such a test will show immediately what his score will be precluding arguments about his physical handicap.

Traffic Rules Complex

Now about traffic rules. All of us are driving at times without knowing what we have a right to do and what is prohibited. We meekly pay our fines and everybody knows that it is no use to argue with the cop, unless one can afford to have a good lawyer afterwards.

What about speed? Everybody knows that danger lies not in speed *per se*, but in speed in reference to conditions. Thirty m.p.h. is safe on a dry concrete, but the same 30 m.p.h. can be extremely dangerous on a wintry, sleety road. What's the sense in indicating a maximum speed? In order to argue in the courts about technicalities?

Do you remember that a few years ago Detroit police collected hundreds of thousands of dollars by enforcing traffic rules, as for instance that speed in downtown sections should be 15 m.p.h. and in turning 7½ m.p.h. plus or minus (how much they do not say)?

We engineers know what the accuracy of a speedometer can be. Result of this campaign, regardless of dollars collected, if that was not the primary object, was that the number of accidents started to grow so alarmingly fast that enforcement had to be called off.

What about the right of way? It is an old joke that Jones had the right of way and that Jones was right. He was dead right. He is dead right now. But it is quite evident that only the right-of-way rule can make an intersection safe. Did you ever hear of any one being arrested for abusing the right of way at an intersection?

What about stop lights? Are there too many or not enough? What about synchronization of these nuisance lights?

Our traffic courts must admit frankly that they know mighty little about preventing accidents.

Erratum

The cutting fluid used in external broaching should be termed "sulphurized" instead of sulphonated, as was printed in the paper by Joseph Geschelin entitled "External Broaching Future Visioned from Study of Current Practice," published in the S.A.E. JOURNAL for February, 1934, p. 70.

Improved Formula for Computing Counterweights of Single-Row and Double-Row Radial Engines

By Prof. A. Coppens
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A "COMPLEMENTARY" term should be added to the usual formula for computing the counterweights of single-row and double-row radial engines, and the complete corresponding formulae are given in the paper under numbers (6), (7), (16), (17), (6′), (7′), (16′), (17′).

The compared effect on engine balance of "complete" and "incomplete" formulas for said counterweight is discussed, with actual figures, in the paper.

I. *Single row radial engines.*—The usual formula for computing the counterweight W of a radial engine is, when this counterweight is supposed to be concentrated at crankradius R from center:

$$W=\underbrace{\left[\frac{CL_1}{L_1}+(n-1)\frac{cl_1}{l}\right]}_{\text{"rotating" weights}}+\underbrace{\frac{\left[np+\frac{CL_2}{L}+(n-1)\frac{cl_2}{l}\right]}{2}}_{\text{"reciprocating" weights}} \qquad (1)$$

in which formula:

n is the number of cylinders

p the weight of one piston

C and c respectively weight of master connecting rod and of link rod

L and l, lengths of both types of connecting rods

L_1 and l_1, distances between piston pin and centre of gravity of connecting rod

L_2 and l_2, distances between big end and centre of gravity of connecting rod

$\frac{CL_1}{L}$ and $\frac{cl_1}{l}$ are respectively the "rotating" weights of both types of connecting rods

$\frac{CL_2}{L}$ and $\frac{cl_2}{l}$ are respectively their "reciprocating" weights.

At the Liege Congress of general mechanics in 1930, the author presented what he thinks a new analytical method for finding out the proper value of counterweights[1], the result of which method is that formula (1) is not correct, as it should be completed by a "complementary term" which is missing in expression (1).

As the use of the complete formula may improve appreciably the balance of radial engines, the author thinks it interesting to expose the question to the Congress.

The trend and the results of the said analytical method can be briefly summarized as follows:

A. Taking first the case of a one-cylinder engine (Fig. 1), its crankcase is supposed to be fastened to some fixed support by bolts, or the like, having certain mechanical characteristics, and which for the present purpose can be represented by springs having stiffnesses h and k and damping coefficients a and b respectively in the horizontal and in the vertical direction.

If then X_o and Z_o are the coordinates of the centre of gravity of crankcase (supposed to be located on centre line of crankshaft for the sake of simplicity), the equations of motion are:

$$\Sigma m\frac{d^2x}{dt^2}=-hX_o-a\frac{dX_o}{dt}$$

$$\Sigma m\frac{d^2z}{dt^2}=-kZ_o-b\frac{dZ_o}{dt}$$

where m, x and z are respectively the mass, the abscissa and the ordinate of all the material points of the engine.

The solving of said equations, of which solving detailed calculations are given in the quoted reference paper, gives for the coordinates X_o and Z_o of the centre of gravity of crankcase, after a sufficient time for allowing the initial

This material was presented as discussion at the Airplane-Engine Session of the International Automotive Engineering Congress, Chicago, Sept., 1933. At this session the following papers were presented:
"Next Decade's Aero Engines Will Be Advanced But Not Radical" by A. H. R. Fedden, published in the S.A.E. Journal, December, 1933, p. 377.
"Engine-Cylinder Flame-Propagation Studied by New Methods" by Kurt Schnauffer, published in the S.A.E. Journal, January, 1934, p. 17.
"Altitude Performance of Aircraft Engines Equipped with Gear-Driven Superchargers" by R. F. Gagg and E. V. Farrar.

[1] Congrès International De Mécanique Générale, Liège 1930: Etablissement d'expressions complètes, exactes et simples de la valeur des contrepoids à appliquer au coudé des machines à piston, en particulier des moteurs d'aviation.—Par Albert Coppens, Professeur à l'Université de Louvain.

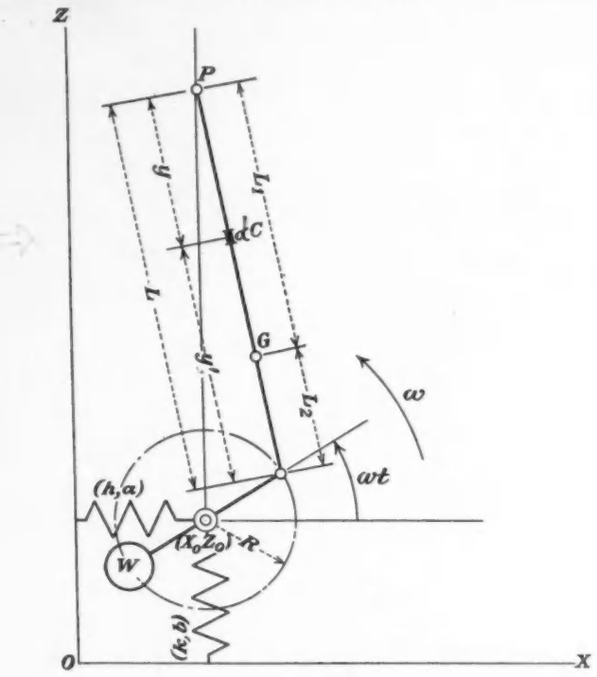

Fig. 1.

transient motion to be damped out:

$$X_o = A - \frac{\left(W - \frac{CL_1}{L}\right)R\omega^2}{g\sqrt{\left(h - \frac{M\omega^2}{g}\right)^2 + a^2\omega^2}} \cos(\omega t + \varphi)$$

$$Z_o = B + \frac{(P+C-W)R\omega^2}{g\sqrt{\left(k - \frac{M\omega^2}{g}\right)^2 + b^2\omega^2}} \sin(\omega t + \psi) + D\cos(2\omega t + X) + E\cos(3\omega t + \xi) + \ldots$$

M is the total weight of engine (including P, C and W); ω the angular velocity of crankshaft and t the time.

In the above values of X_o and Z_o, A and B are constant terms depending on the peculiar location of OX and OZ relatively to the engine; ψ, X, ξ, ... are angles whose values need not to be given here, and D, E . . . . stand for expressions in which the value W of counterweight does not come in.

Similar, but somewhat more complicated expressions are found for components F_x and F_z of the force exerted by fastening bolts on crankcase, of which expressions we write down here only the terms in ωt ("fundamental motion"):

$$Fx = +\frac{\left(W - \frac{CL_1}{L}\right)R\omega^2}{g}\left[\frac{h\cos(\omega t+\varphi) - a\omega\sin(\omega t+\varphi)}{\sqrt{\left(h - \frac{M\omega^2}{g}\right)^2 + a^2\omega^2}}\right]$$

$$Fz = -\frac{(P+C-W)R\omega^2}{g}\left[\frac{k\sin(\omega t+\psi) + b\omega\cos(\omega t+\psi)}{\sqrt{\left(k - \frac{M\omega^2}{g}\right)^2 + b^2\omega^2}}\right]$$

and whose terms in $2\omega t$, $3\omega t$, etc. do not include at all the value W of counterweight.

Accordingly, only the fundamental motion in ωt is affected by the value W of the counterweight. This fundamental motion is an elliptical motion. Displacements X_o and Z_o and components F_x and F_z of force cannot be annulated simultaneously in the horizontal and in the vertical direction, as it is impossible to make at the same time:

$$W = \frac{CL_1}{L}$$

which is smaller than C, and $W = P + C$ which is larger than C.

The exact values of h and k and of a and b are usually unknown: but it is sufficient to suppose that $h = k$ and $a = b$ to be able to ascertain which is the best value to be selected for W. In this case the fundamental motion in ωt is:

$$X_o = -\frac{\left(W - \frac{CL_1}{L}\right)R\omega^2}{g\sqrt{\left(h - \frac{M\omega^2}{g}\right)^2 + a^2\omega^2}} \cos(\omega t+\varphi)$$

$$Z_o = +\frac{(P+C-W)R\omega^2}{g\sqrt{\left(h - \frac{M\omega^2}{g}\right)^2 + a^2\omega^2}} \sin(\omega t+\varphi)$$

$$Fx = +\frac{\left(W - \frac{CL_1}{L}\right)R\omega^2}{g}\left[\frac{h\cos(\omega t+\varphi) - a\omega\sin(\omega t+\varphi)}{\sqrt{\left(h - \frac{M\omega^2}{g}\right)^2 + a^2\omega^2}}\right]$$

$$Fz = -\frac{(P+C-W)R\omega^2}{g}\left[\frac{h\sin(\omega t+\varphi) + a\omega\cos(\omega t+\varphi)}{\sqrt{\left(h - \frac{M\omega^2}{g}\right)^2 + a^2\omega^2}}\right]$$

The maximum values Xm, Zm, Fxm and Fzm of Xo, Zo, Fx and Fz are:

$$Xm = \frac{\left(W - \frac{CL_1}{L}\right)R\omega^2}{g\sqrt{\left(h - \frac{M\omega^2}{g}\right)^2 + a^2\omega^2}}$$

$$Zm = \frac{(P+C-W)R\omega^2}{g\sqrt{\left(h - \frac{M\omega^2}{g}\right)^2 + a^2\omega^2}}$$

$$Fxm = \frac{\left(W - \frac{CL_1}{L}\right)R\omega^2\sqrt{h^2+a^2\omega^2}}{g\sqrt{\left(h - \frac{M\omega^2}{g}\right)^2 + a^2\omega^2}}$$

$$Fzm = \frac{(P+C-W)R\omega^2\sqrt{h^2+a^2\omega^2}}{g\sqrt{\left(h - \frac{M\omega^2}{g}\right)^2 + a^2\omega^2}}$$

When now W is made to increase from zero upwards, X_o, Z_o, F_x, and F_z vary according to Fig. 2.

Now, the best value for the counterweight W will be the value which will make minimum the greatest of either X_m and Z_m, and of either F_{xm} and F_{zm}, i.e. the value which makes $X_m = Z_m$ and $F_{xm} = F_{zm}$.

This best value of W will thus be given by:

$$\left(W - \frac{CL_1}{L}\right) = [P+C-W$$

hence:

$$W = \frac{P + C + \frac{CL_1}{L}}{2} = \frac{CL_1}{L} + \frac{P + \frac{CL_2}{L}}{2}$$

If this best value is given to W, then the centre of gravity

of crankcase describes, according to Fig. 2, a circle, with angular velocity ω in the direction opposite to rotation of crankshaft; and, in the same way, resulting force F, whose components are F_x and F_z, has a constant value and turns round in a direction opposite to crankshaft rotation.

The expression

$$W=\frac{CL_1}{L}+\frac{P+\frac{CL_2}{L}}{2}$$

is the usual form; "rotating" weights plus half of "reciprocating" weights; but the expression

$$W=\frac{P+C+\frac{CL_1}{L}}{2}$$

is a form which falls better in with the present demonstration: mean value between what is required for balancing vertical motion and what is required for balancing horizontal motion of crankcase. This best value of W, it is to be recalled, is completely independent of the actual values of stiffnesses h and k and of damping coefficients a and b, if only $h = k$ and $a = b$. It is to be pointed out, too, that in the present analytical method no assumption whatever has been made on the way of replacing the total weight C of connecting rod by a "rotating" weight and a "reciprocating" weight: in the demonstration as given in the quoted reference, expression $\frac{CL_1}{L}$ for instance is found in the following way (Fig. 1):

$$\frac{1}{L}\int ydC=\frac{CL_1}{L}$$

where y is the distance between elementary weight dC and piston pin.

B. Going over now to the single row radial engine, Fig. 3, it can be taken for granted that in all the practical cases, the method of fastening engine to plane, by a crown of bolts

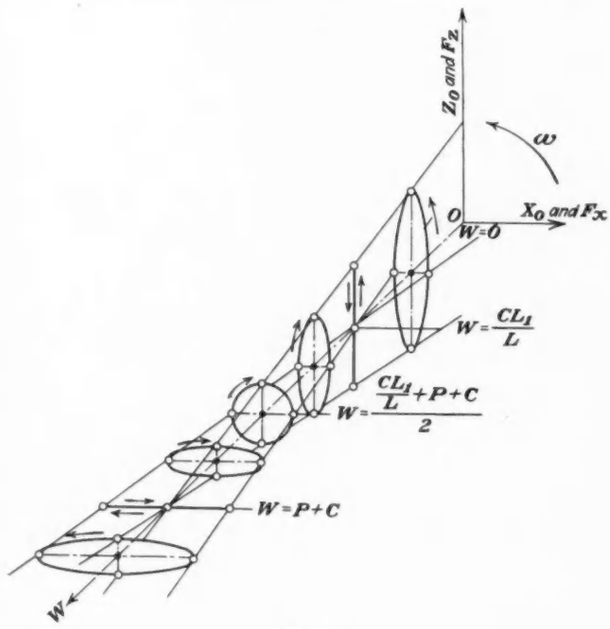

Fig. 2.

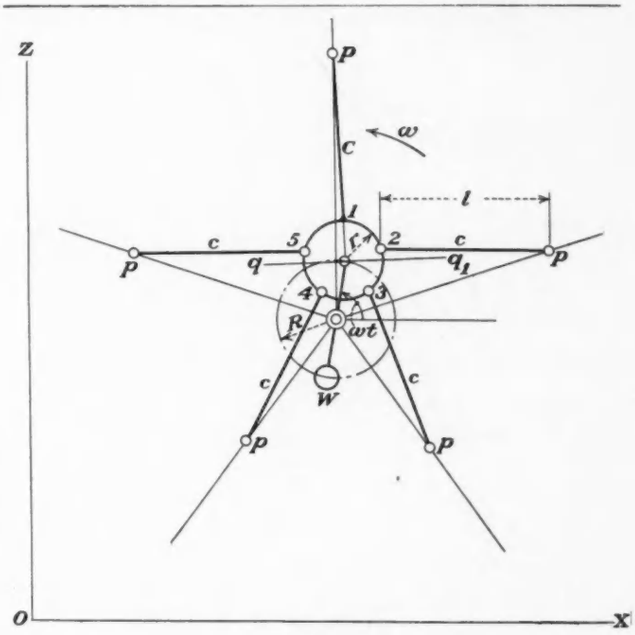

Fig. 3.

equally or about equally spaced, is such that really $h = k$ and $a = b$. As the results are, in this case, independent of the actual values of $h = k$ and $a = b$, we choose, for this stiffness and damping coefficient, the values which make the calculations most easy. These values are: (a) either $h = k = \infty$ and $a = b = o$: (infinitely stiff bolts: damping coefficients equal to zero). In this case the calculations given in quoted reference paper (calculations which are very complicated, notwithstanding these simplifying chosen values) give for the fundamental motion in ωt, the only one which is affected by the value W of counterweight:

$$Fx=-\frac{1}{g}\Big[\frac{CL_1}{L}-\frac{cl_1}{l}+\frac{cl_1}{l}\frac{r}{L}+\frac{n}{2}\Big(\frac{cl_1}{l}+p+c\Big)-W\Big]R\omega^2\cos\omega t \qquad (2)$$

($X_o=0$, $Z_o=0$)

$$Fz=-\frac{1}{g}\Big[C-c+\frac{n}{2}\Big(\frac{cl_1}{l}+p+c\Big)-W\Big]R\omega^2\sin\omega t \qquad (3)$$

These expressions mean that the centre of gravity of crankcase remains motionless and that the forces F_x and F_z vary according to an elliptic law; r is the distance between big end of link rod and crankpin as shown on Fig. 3. (b) or $h = k = o$ and $a = b = o$ (infinitely flexible bolts and damping coefficients equal to zero: this is the case of the engine supposed to be isolated in space). The fundamental motion in ωt is:

$$X_o=\frac{1}{M}\Big[\frac{CL_1}{L}-\frac{cl_1}{l}+\frac{cl_1}{l}\frac{r}{L}+\frac{n}{2}\Big(\frac{cl_1}{l}+p+c\Big)-W\Big]R\cos\omega t \qquad (4)$$

$$Z_o=\frac{1}{M}\Big[C-c+\frac{n}{2}\Big(\frac{cl_1}{l}+p+c\Big)-W\Big]R\sin\omega t \qquad (5)$$

($Fx=0$, $Fz=0$)

The center of gravity of crankcase describes an ellipse. In either supposition, the best value for the counterweight W will be the value which renders the maximum value of F_x equal to the maximum value of F_z or the maximum value of

X_o equal to the maximum value of Z_o. Either condition gives:

$$\left[\frac{CL_1}{L}-\frac{cl_1}{l}+\frac{cl_1}{l}\frac{r}{L}+\frac{n}{2}\left(\frac{cl_1}{l}+p+c\right)-W\right]=$$

$$-\left[C-c+\frac{n}{2}\left(\frac{cl_1}{l}+p+c\right)-W\right] \quad \text{hence:}$$

$$W=\left[\frac{CL_1}{L}+(n-1)\frac{cl_1}{l}\right]+\frac{\left[np+\frac{CL_2}{L}+(n-1)\frac{cl_2}{l}\right]}{2}+\frac{cl_1}{2l}\frac{r}{L} \quad (6)$$

i.e., the counterweight should include, besides the "rotating" weights and half the "reciprocating" weights of formula (1) a complementary term $\frac{cl_1}{2l}\frac{r}{L}$ without which the expression of W is incomplete and not correct.

The following form of expression (6):

$$W=\frac{np+C+(n-1)c+\frac{CL_1}{L}+(n-1)\frac{cl_1}{l}+\frac{cl_1}{l}\frac{r}{L}}{2} \quad (7)$$

mean value between what is required for balancing "total" weights and what is required for balancing "rotating" weights, falls in better with the present demonstration.

Remark 1. It is interesting to know which is the compared effect on forces and displacements of using the incomplete or the complete expression of counterweight.

(a) By replacing in formulas (2) and (3), (4) and (5), W by its incomplete expression (1):

$$Fx=+\frac{1}{g}\left[\frac{CL_2}{2L}-\frac{cl_2}{2l}-\frac{cl_1}{l}\frac{r}{L}\right]R\omega^2\cos\omega t \quad (8)$$

$$Fz=-\frac{1}{g}\left[\frac{CL_2}{2L}-\frac{cl_2}{2l}\right]R\omega^2\sin\omega t \quad (9)$$

$$X_o=-\frac{1}{M}\left[\frac{CL_2}{2L}-\frac{cl_2}{2l}-\frac{cl_1}{l}\frac{r}{L}\right]R\cos\omega t \quad (10)$$

$$Z_o=+\frac{1}{M}\left[\frac{CL_2}{2L}-\frac{cl_2}{2l}\right]R\sin\omega t \quad (11)$$

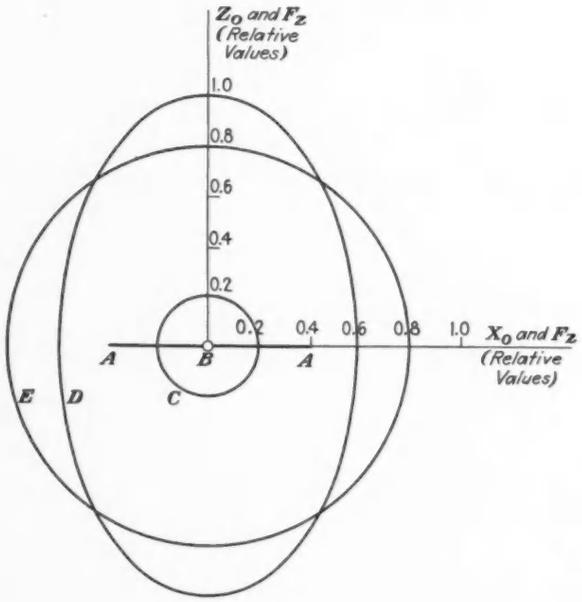

Fig. 4.

The motion is elliptic, with vertical components of forces and displacements greater than horizontal components. Direction of rotation opposite to rotation of crankshaft.

(b) By replacing in the same formula W by its complete expression (6) or (7):

$$Fx=+\frac{1}{g}\left[\frac{CL_2}{2L}-\frac{cl_2}{2l}-\frac{cl_1}{2l}\frac{r}{L}\right]R\omega^2\cos\omega t \quad (12)$$

$$Fz=-\frac{1}{g}\left[\frac{CL_2}{2L}-\frac{cl_2}{2l}-\frac{cl_1}{2l}\frac{r}{L}\right]R\omega^2\sin\omega t \quad (13)$$

$$X_o=-\frac{1}{M}\left[\frac{CL_2}{2L}-\frac{cl_2}{2l}-\frac{cl_1}{2l}\frac{r}{L}\right]R\cos\omega t \quad (14)$$

$$Z_o=+\frac{1}{M}\left[\frac{CL_2}{2L}-\frac{cl_2}{2l}-\frac{cl_1}{2l}\frac{r}{L}\right]R\sin\omega t \quad (15)$$

The motion is, as it should be, circular, with vertical components of forces and displacements equal to horizontal components, and direction of rotation opposite to rotation of crankshaft.

Remark 2. In most cases of actual engines, the weight of the various pistons and link rods are not exactly the same and the values of r may even be made purposely different in order to insure the same compression ratio in all the cylinders. As shown in the quoted paper, the generalized expression of counterweight, taking account of these differences, is:

$$W=\left[\frac{CL_1}{L}+\Sigma\frac{cl_1}{l}\right]+\frac{\left[\Sigma p+\frac{CL_2}{L}+\Sigma\frac{cl_2}{l}\right]}{2}+\frac{\Sigma\frac{cl_1}{l}r}{2(n-1)L} \quad (16)$$

corresponding to expression (6)

or

$$W=\frac{\Sigma p+C+\Sigma c+\frac{CL_1}{L}+\Sigma\frac{cl_1}{l}+\frac{\Sigma\frac{cl_1}{l}r}{(n-1)L}}{2} \quad (17)$$

corresponding to expression (7)

II. Double row radial engines.—As is well known, double-row radial engines of $2n$ cylinders require two counterweights, one for each row. If d is the distance between the planes of both cylinder rows, and d^1 the distance between the planes of both counterweights, the complete expression for the value W^1 of each said counterweight should be:

$$W^1=\frac{d}{d^1}\left[\left(\frac{CL_1}{L}+(n-1)\frac{cl_1}{l}\right)+\frac{np+\frac{CL_2}{L}+(n-1)\frac{cl_2}{l}}{2}+\frac{cl_1}{2l}\frac{r}{L}\right]$$

or:

$$W^1=\frac{d}{d^1}\left[\frac{np+C+(n-1)c+\frac{CL_1}{L}+(n-1)\frac{cl_1}{l}+\frac{cl_1}{l}\frac{r}{L}}{2}\right] \quad (6')$$

$$(7')$$

and the generalized expressions should be:

$$W^1=\frac{d}{d^1}\left[\left(\frac{CL_1}{L}+\Sigma\frac{cl_1}{l}\right)+\frac{\Sigma p+\frac{CL_2}{L}+\Sigma\frac{cl_2}{l}}{2}+\frac{\Sigma\frac{cl_1}{l}r}{2(n-1)L}\right] \quad (16')$$

or

$$W^1=\frac{d}{d^1}\left[\frac{\Sigma p+C+\Sigma c+\frac{CL_1}{L}+\Sigma\frac{cl_1}{l}+\frac{\Sigma\frac{cl_1}{l}r}{(n-1)L}}{2}\right] \qquad (17')$$

instead of the incomplete formulae in which the term:

$$\frac{d}{d^1}\ \frac{cl_1}{2l}\ \frac{r}{L} \text{ or } \frac{d}{d^1}\ \frac{\Sigma\frac{cl_1}{l}r}{2(n-1)L}$$

is missing.

Relative Importance of the Complementary Term

In order to discuss this importance, a concrete example should be taken: for instance, the following round figures which are approximately those of a 450 hp. single-row nine-cylinder engine:

$p=4$ lb. $\quad \frac{CL_1}{L}=12$ lb. $\quad \frac{cl_1}{l}=1$ lb.

$\frac{CL_2}{L}=2$ lb. $\quad \frac{cl_2}{l}=1$ lb.

$C=\overline{14 \text{ lb.}}$ $\quad c=\overline{2 \text{ lb.}}$

$R=4$ in. $\quad L=15$ in. $\quad r=3$ in. $\quad l=12$ in.

$M=800$ lb.

The incomplete formula (1) gives	$W=43.0$ lb.
The complementary term is	0.1 lb.
The complete formula (6) or (7) gives	$W=43.1$ lb.

The complementary term is only 0.23 per cent of W, which value is so small that is seems at first sight to be negligible. But in order to ascertain what its real bearing is on the balance of the engine, let the corresponding numerical values of F_x, and F_z, X_o, Z_o be calculated. As ω figures in expressions of F_x and F_z let us take $N = 1910$ revs, $\omega = 200$. These numerical values replaced in formulas (2), (3), (4) and (5) give:

$$Fx=-414\ (42.7-W)\cos\omega t$$
$$Fz=-414\ (43.5-W)\sin\omega t$$
$$X_o=+0.005\ (42.7—W)\cos\omega t$$
$$Z_o=+0.005\ (43.5-W)\sin\omega t$$

The "incomplete" value of counterweight: $W=$ 43.0 lb. will give:

$Fx=+124.2\cos\omega t$ $\quad X_o=-0.0015\cos\omega t$

$Fz=-207.2\sin\omega t$ $\quad Z_o=+0.0025\sin\omega t$

which motion is represented in Fig. 4 by the ellipse D. whereas the "complete" value $W =$ 43.1 lb. will give:

$Fx=+165.6\cos\omega t$ $\quad X_o=-0.002\cos\omega t$

$Fz=-165.6\sin\omega t$ $\quad Z_o=+0.002\sin\omega t$

which motion is represented in Fig. 4, by the circle E. Accordingly, increasing the balance weight from 43.0 lb. to 43.1 lb. or only by 0.23 per cent, reduces the maximum force between crankcase and supporting frame from 207.0 lb. to 165.6 lb. and the maximum displacement of crankcase from 0.0025 in. to 0.002 in. i.e., from 1 to 0.8 or by 20 per cent. Same improvement of 20 per cent would remain whatever be the stiffness and damping coefficient of fastening bolts; and Fig. 5 shows clearly that this best balance of engine occurs quite "sharply" at 43.1 counterweight.

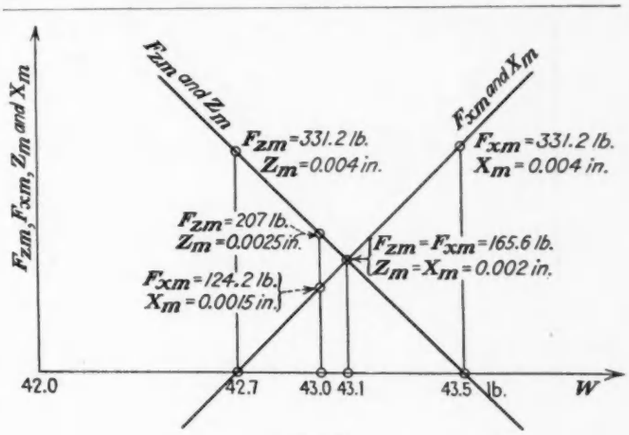

Fig. 5.

Remark 1. The forces F_x and F_z represent also the pressures between crankshaft and mean bearings, when said crankshaft is assumed to be perfectly rigid; so that for reducing said pressures to a minimum, the complete formula, instead of the incomplete formula, should be used for calculation of counterweight.

Remark 2. The improvement of 20 per cent brought about by using the complementary term will be indeed very hard to detect by actual experiments, as the expressions of X_o and Z_o show that, even in the case of infinitely flexible fastening bolts, displacements of center of gravity of crankshaft would vary between the very small values of 2.5 to 2.0 thousandths of an inch. These displacements could be, of course, made greater by spring-supporting the crankcase and running the crankshaft at the critical speed corresponding to the flexibility of said springs; but they would remain in any case very small; so that one has really to rely on the correct calculations for preferring the complete formula to the incomplete formula.

Remark 3: The simplified theory from which has probably been evolved the incomplete formula (1) would lead naturally to the conclusion that if it was possible to reduce the "reciprocating weight $\frac{CL_2}{L}$ of the master connecting rod to the value of the "reciprocating" weight $\frac{cl_2}{l}$ of the link rods, then the proper counterweight would completely balance out the fundamental forces and displacements in ωt of the engine.

But this conclusion would be false: for it can be shown in the following way that if $\frac{CL_2}{L}$ was made equal to $\frac{cl_2}{l}$, neither the "incomplete" nor the "complete" counterweight would balance completely the engine.

The expressions (8) (9) (10) and (11) show that with the "incomplete" counterweight, condition $\frac{CL_2}{L}=\frac{cl_2}{l}$ would balance completely the vertical motion, but not the horizontal one:

$$Fx=-\frac{1}{g}\ \frac{cl_1}{l}\ \frac{r}{L}R\omega^2\cos\omega t \qquad X_o=+\frac{1}{M}\ \frac{cl_1}{l}\ \frac{r}{L}R\cos\omega t$$
$$Fz=0 \qquad Z_o=0$$

or, with the numerical figures of our actual example, in which the weight of the master connecting rod is reduced in

the following way:

$$\frac{CL_1}{L}=12 \text{ lb.}$$

$$\frac{CL_2}{L}=1 \text{ lb.} = \frac{cl_2}{l} \qquad W=42.5$$

$$C=13 \text{ lb.}$$

$$Fx=-82.8\cos\omega t \qquad X_o=+0.001\cos\omega t$$

$$Fz=0 \qquad Z_o=0$$

which motion is represented by the *horizontal line AA* drawn to scale in Fig. 4.

In the same way, the expressions (12) (13) (14) and (15) show that, even with the complete counterweight, condition $\frac{CL_2}{L}=\frac{cl_2}{l}$ would not completely balance the engine:

$$Fx=-\frac{1}{g}\frac{cl_1}{2l}\frac{r}{L}R\omega^2\cos\omega t \qquad X_o=+\frac{1}{M}\frac{cl_1}{l}\frac{r}{L}R\cos\omega t$$

$$Fz=+\frac{1}{g}\frac{cl_1}{2l}\frac{r}{L}R\omega^2\sin\omega t \qquad Z_o=-\frac{1}{M}\frac{cl_1}{2l}\frac{r}{L}R\sin\omega t$$

or, with our numerical figures:

$$W=42.6$$

$$Fx=-41.4\cos\omega t \qquad X_o=+0.0005\cos\omega t$$

$$Fz=+41.4\sin\omega t \qquad Z_o=-0.0005\sin\omega t$$

which motion is represented by the small circle C drawn to scale in Fig. 4.

In order to enable the "complete" counterweight to balance completely the fundamental motion in ωt, i.e., to make F_x, F_z, X_o and $Z_o = o$ formulae (12) (13) (14) (15) show clearly that it is necessary to reduce the "reciprocating" weight $\frac{CL_2}{L}$ only to the value:

$$\frac{CL_2}{L}=\frac{cl_2}{l}+\frac{cl_1}{l}\frac{2}{L},$$

or with our numerical figures, to: 1.0 + 0.2 = 1.2 lb. so that the particulars of the master connecting rod should be

$$\frac{CL_1}{L}=12.0 \text{ lb.}$$

$$\frac{CL_2}{L}=1.2 \text{ lb.}$$

$$C=13.2 \text{ lb.}$$

The correct value of counterweight should be then, by applying "complete" formula (6) or (7):

$$W=42.7$$

and would reduce completely to zero both forces and displacements in ωt of the engine. (point B of Fig. 4.)

Conclusion

In the manufacturing of aero-engines as well as of any other part of aircraft equipment, everything should be, as is well known, the best possible; the only departure to be made from this policy is when the additional cost is absolutely prohibitive. In the present case, the complete formula is the better one; and the cost of calculating, for each engine, or for each type of engine, the expression:

$$\frac{\Sigma\frac{cl_1}{l}r}{2(n-1)L}$$

and of adding it to the main part of the value of the counterweight is only a mere trifle; so that there is no reason whatever not to replace the incomplete formula by the complete formula.

Reproducibility of Test for Knock Characteristics of Fuels

A REPORT of the cooperative work by the Subcommittee on Methods of Measuring Detonation of the Cooperative Fuel Research Steering Committee, recently submitted by Harry F. Huf, includes the following information:

A Tentative Method of Test for Knock Characteristics of Motor Fuels (A.S.T.M. Designation: D357-33T) was accepted for publication by A.S.T.M. Committee E-10 on Standards on March 7, 1933. In Section 16 of the tentative method this statement appears: "Results obtained by this procedure with different C.F.R. engines and in different laboratories should differ by not more than two octane numbers." However, experience in use of the method over a period of time indicated that errors or disagreements between laboratories sometimes exceeded two octane numbers. The work reported here was instigated (a) to gather data on the reproducibility of the method and (b) to determine by analysis of data the probable causes of errors in excess of stated limits.

The laboratories cooperating in the tests were members of either the Cooperative Fuel Research Steering Committee or its Subcommittee on Methods of Measuring Detonation, and included the following:

The Atlantic Refining Co.; Bureau of Standards; Continental Oil Co.; Empire Oil and Refining Co.; Ethyl Gasoline Corp.; General Motors Corp.; Phillips Petroleum Co.; Shell Petroleum Co.; Sinclair Refining Co.; Socony-Vacuum Co.; Standard Oil Development Co.; Standard Oil Co. of California; Standard Oil Co. of Indiana; Sun Oil Co.; The Texas Co.; Tidewater Oil Co.; Universal Oil Products Co.; Waukesha Motor Co.; Yale University.

Ten samples were distributed by ten of the cooperating laboratories.

Conclusions reached as a result of the tests were:

1. When A.S.T.M. knock test equipment is properly adjusted, octane numbers determined in accordance with the A.S.T.M. method differ by less than two octane numbers, and the average error is less than 0.5 octane number.

2. Errors greater than 1.0 octane number are usually caused by improper engine conditions.

Recommendations were:

1. The Waukesha Motor Co. is endorsed as a referee and control laboratory, and it is urged that they be consulted in cases where a persistent error in one direction is encountered.

2. It is recommended that cooperative effort be aimed at making available for general use a check fuel, sensitive to engine adjustment, reproducible, and chemically stable, to enable operators to check the condition of their equipment.